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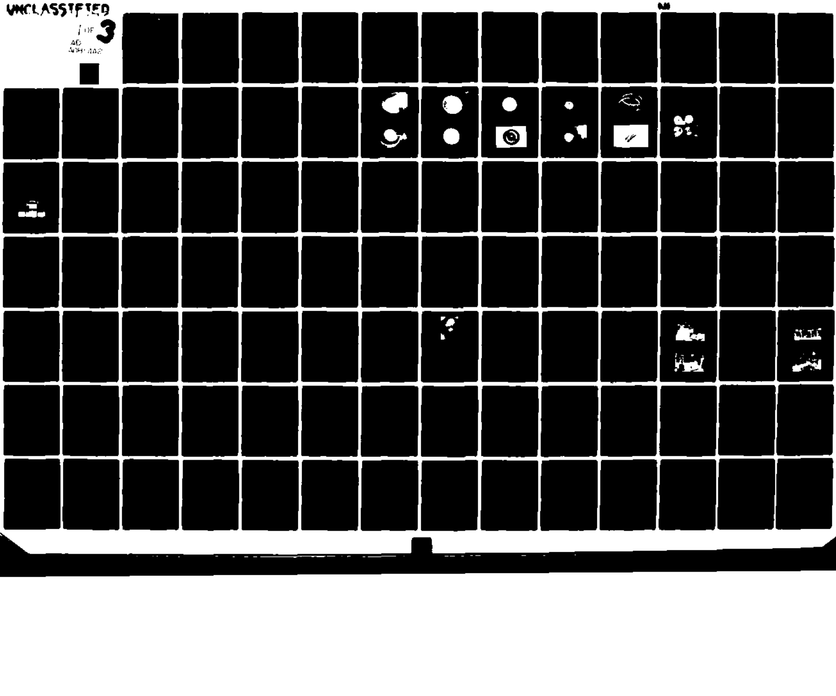

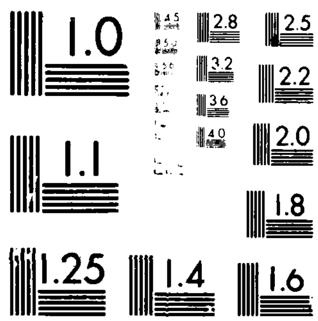

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FINAL REPORT

CONTRACT # DAAK70-78-C-0031

TURBOCHARGING OF SMALL INTERNAL COMBUSTION ENGINE AS A MEANS OF IMPROVING ENGINE/APPLICATION SYSTEM FUEL ECONOMY

PREPARED BY

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TABLE OF CONTENTS

SECTION

I. Summary

II. Preface

III. Introduction

- A. Purpose
- B. Background
  - 1. Summary of turbocharger design
  - 2. Simulated rotor test rig
- C. Program Breakdown and Scope of Work

IV. Investigation

- A. Manufacture turbocharger
- B. Bench Testing
  - 1. oil/wick/wick-shaft interface tests
  - 2. complete turbocharger testing
- C. Develop Mathematical Models
- D. Select Engine
- E. Engine Performance Tests
- F. Predict Fuel Consumption & Emissions

V. Discussion

- A. Manufacture Turbocharger
- B. Bench Tests
- C. Engine Performance Tests
- D. Predict Fuel Consumption

VI. Conclusions

A. Manufacture Turbochargers

B. Bench Tests

C. Mathematical Models

D. Engine Performance Tests

E. Predict Fuel Consumption

F. General Conclusion

VII. Recommendations

Appendix: "A" Simulated Rotor

"B" Oil/Wick/Wick-Shaft Interface Tests

"C" Test Facility

"D" FINAL REPORT Southwest Research Institute

LIST OF ILLUSTRATIONS

FIGURE #		Page
1	Turbocharger	8
2	Compressor Housing	13
3	Turbine Housing	13
4	Compressor Backwall	14
5	Turbine Backwall	14
6	Turbine Backwall With Heatshield and Control Levers	15
7	Turbine Backwall and Nozzles	15
8	Compressor Wheel Casting	16
9	Turbine Wheel Casting	16
10	Turbocharger Clamps	17
11	Oil Wicks	17
12	Turbocharger Piece Parts & Sub Assemblies	18
13	Oil Wick Test Rig	21
14	Compressor Map-Vaned Design Flow	24
15	Compressor Map-Vaned High Flow	25
16	Compressor Map-Vaned Low Flow	26
17	Compressor Map-Vaneless High Flow	27
18	Specific Fuel Consumption Vs RP, and BHP	32
19	Emission and Fuel Economy Over 13 Mode Federal Diesel Emission Cycle	33

LIST OF ILLUSTRATIONS cont.

FIGURE #		Page
20	Bar Plots of Smoke Test Results	34
21	Model Estimation of Fuel Economy For Various N/V Ratios	36
22	Compressor Map - Turbocharger Received on John Deere Engine	40
23	Measured Turbine Efficiency	42
24	Theoretical and Actual Air Flow Plotted on Compressor Map	46
A1	Simulated Rotor Test Rig	56
A2	Simulated Rotor Test RPM Vs. Time	57
C1	Cummins NH250 Diesel Engine	60
C2	Exhaust Plenum 80 Gallon Tank	60
C3	Compressor Inlet Ducting	62
C4	Compressor Exit Ducting	62
C5	Test Cell Facility	66

I. SUMMARY

This report presents the results of prototype manufacturing, rig testing, application, and engine testing of a small advanced technology turbocharger. The turbocharger features variable turbine nozzles, ball bearings supported rotor system, self contained lube system and a broad operating range compressor. The purpose of the work was to show the potential benefits of the subject turbocharger in enhancing specific fuel consumption, emissions, and transient response of a diesel engine. The work was accomplished through laboratory testing of hardware and subsequent mathematical duty cycle simulation using the acquired data.

The proposed turbocharger was manufactured and successfully run on a turbocharger test rig. Compressor maps were generated for several compressor trims with vaned and vaneless diffusers. A turbocharger was successfully run for 53 hours on a John Deere, 239 cubic inch, four cylinder, diesel engine. Fuel consumption and emissions data were obtained for this engine as well as the "as received" turbocharged engine and the engine with no turbocharger.

Best specific fuel consumption was equal to or better than the "as received" turbocharged engine. In general, the fuel consumption was improved at all conditions except medium speed, medium to high

load where the original turbocharger was apparently optimized. Emissions were responsive to turbine nozzle position. Closed nozzles (producing higher turbocharger speeds and intake manifold pressures) produced greater NO_2 and less CO, hydrocarbons and smoke than the baseline "as received" turbocharged engine. Open nozzles produced the opposite results. Transient testing was inconclusive.

Test data showed that the compressor was not well matched to the engine. Further, the exhaust temperatures were much lower than the initially assumed (1190°F max. versus 1600°F) design point. The turbocharger was therefore rather poorly matched to the engine.

Data reduction also showed that more heat was being transfered from the turbine to the compressor than was anticipated. This resulted in reduced intake manifold densities (than theoretically possible with no heat transfer) and therefore, reduced air mass flow.

The extremes of nozzle travel (generally $\pm$ 10 degrees) did not seem to produce the extremes of potential improvement.

The general conclusion reached is that, in spite of the poor aerodynamic match and the adverse heat transfer condition, an advanced turbocharger with variable area turbine nozzles, a broad operating range compressor and very low loss anti-friction bearings can produce lower specific fuel consumption, can "flatten" the sfc versus engine speed (at constant horsepower) characteristics and can be an effective control variable for emissions. A fully developed turbocharger, appropriately matched, will give the engine designer a new tool, heretofore not available, for matching a diesel powerplant to a specific requirement while optimizing fuel consumption and emissions. A more exhaustive effort, utilizing a better matched turbocharger, is required to better define the potentials.

II. PREFACE

This work was authorized by contract DAAK70-78-C0031 administered by the Electromechanical Division of the Mobility Equipment Research and Development Command, Ft. Belvoir, Virginia. The Contracting Officer was John A. Gabby. The Contracting Officers' Technical Representative was Paul Arnold. Robert Ware contributed valuable reviews and suggestions. The effort was funded through the U. S. Army Advanced Concepts Team, whose Executive Director is Dr. Charles Church, as a result of an unsolicited proposal. Dr. Church provided considerable overall guidance to the effort.

Dr. Koneru Tataiah of Southwest Research Institute, San Antonio, Texas managed and supervised the engine test portion of the effort as well as formulated and programmed the mathematical model. This work was conducted in the Department of Engine and Vehicle Research, Charles Wood, Director. Mr. Wood contributed much in guidance and specific suggestions.

It should be noted here that Southwest Research Institute wrote a final report on their efforts and it is attached as Appendix "D". For those areas that were predominately Southwest Research Institute work, the objectives and basic results will be presented with reference to their report for the particulars.

III. INTRODUCTION

A. Purpose

The purpose of this effort was to demonstrate the technical feasibility of using an advanced design turbocharger (featuring variable area turbine nozzles (VATN), a ball bearing supported rotor system, a self contained lubrication system and a broad operating range compressor) to improve specific fuel consumption, emissions, and transient response of a diesel engine.

B. Background

1. summary of turbocharger design -

Aerodyne recognized the need for an improvement in the state-of-the-art of small turbochargers, particularly in the following areas:

* mechanical efficiency
* control
* bearing life
* operating range

A design concept evolved that held promise for improvements in the targeted areas. The turbocharger design concept features variable area turbine nozzles (VATN), ball bearing supported rotor system, a self contained lubrication system and a broad operating range compressor.

The broad operating range compressor, used in conjunction with the VATN, allows exceptionally broad ranges of efficiently controlled operation with respect to engine speed and boost pressure. The VATN also provides additional turbine power output for improved transient response. Additionally, the low friction ball bearings provide dramatic improvements in mechanical efficiency - reducing the steady state turbine power requirement as well as enhancing transient response. The ball bearings provide a relatively "stiff" rotor system which allows reduced blade tip running clearances - thereby improving compressor and turbine efficiency.

Additionally, the rotor system is overhung placing the bearings in the cool environment of the compressor inlet. This allows a self contained, wick fed lubrication system with the following benefits:

* no seals are required - any excess oil (which is minimal) is simply passed through the engine

* any shaft orientation can be run (including vertical)

* engine oil and associated plumbing is not required - contaminated engine oil or the lack of engine oil is the primary cause of bearing and seal failures in present turbochargers

* the bearing system is considerably less complex than journal/thrust bearing systems

A detailed design of a specific turbocharger was completed for a spark ignition engine. The design point was chosen for what Aerodyne judged to be future typical automotive requirements. The aerodynamic design point of the turbocharger was: a corrected flow of 200 CFM ($Q/\sqrt{\frac{T1}{519}}$) at a compressor pressure ratio of 2.3 (R_c) (vaned diffuser) and a turbine inlet temperature of 2060^{o}R at a fuel/air ratio of .067 with compressor inlet loss of 1 inch of mercury and a turbine discharge loss of 6 inches of mercury.

A cross-section of this turbocharger is shown in Figure 1.

2. simulated rotor test rig

In order to verify the rotor/bearing/lube system design approach a simulated rotor rig was constructed and run. The details of this effort are presented in Appendix "A".

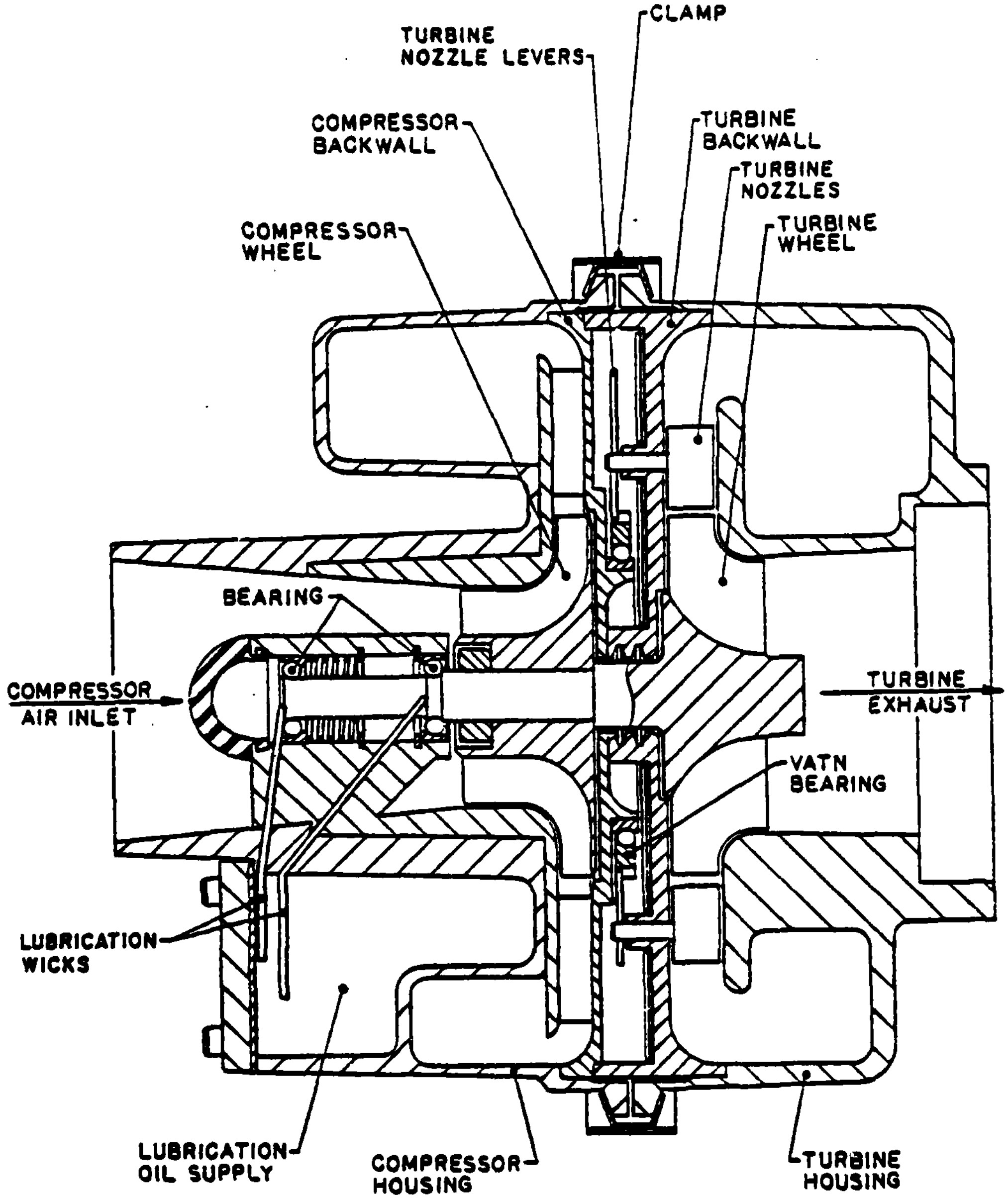


FIGURE 1 - TURBOCHARGER

C. Program Breakdown and Scope of Work

This report covers the program outlined below as well as the conclusions drawn from the results and the recommendations.

The program.was broken down into the following major areas:

1. manufacture turbochargers
2. conduct bench tests to characterize the turbocharger's operation
3. develop a mathematical model to predict fuel consumption and emissions for small turbocharged diesel engines for a selected automotive driving cycle
4. select a commercially available test engine and apply the turbocharger to it
5. conduct engine tests to define operating characteristics at various VATN settings
6. predict fuel consumption and emissions, using the developed mathematical model and actual engine test data

IV. INVESTIGATION

A. Manufacture Turbochargers

The objectives of this effort was to make provision for the materials, tooling, processing, and assembly necessary for the manufacture of turbochargers.

A brief description of the turbocharger follows:

The rotor consists of an overhung back-to-back compressor/turbine arrangement with the bearings located in the relatively cool compressor inlet. The bearing is a full complement instrument ball bearing with the inner raceway being an integral part of the shaft. Slinger ramps are provided, adjacent to the inner raceways, on which wicks contact the shaft. These wicks, which are immersed in a reservoir of oil, continually "write" a film of oil on the slinger ramps during shaft rotation. The oil reservoir is integrally cast with the compressor housing. Centrifugal force then causes the oil to be "slung" from the sharp intersection of the slinger ramp with raceway onto the balls and outer raceway. Thus, a miniscule flow of very clean oil is provided to the bearings during operation. At rest no flow exists. A compression spring preloads the bearings. The compressor wheel is captured axially and driven by an interference fit sleeve with driving lugs that engage the compressor wheel. No seals are required or used in the bearing system design.

A constant velocity scroll with a single discharge is used to collect and deliver compressor air. A similar type scroll is used for the turbine to prepare the gases for the turbine nozzles.

The VATN actuating mechanism is located in the air space between the compressor and turbine and consists of:

* stamped sheet metal levers with a "D" shaped indexing hole for mounting on the turbine nozzle vane trunnions and engagement means for locating in the coordinating ring. One of the levers extends radially outward, having provision for attaching a rod leading to an actuator.

* the VATN bearing, which is a large bore ball bearing with the outer race being the coordinating ring with slots for engagement of the levers.

An asbestos heat shield is provided between the turbine backwall and the VATN mechanism. The heat shield and air space provide the heat transfer barrier between the turbine and compressor.

The four major structural members are clamped axially by a single 'V' clamp and are piloted such that thermal expansion causes increased radial interference.

Turbochargers were successfully manufactured. Casting tooling was procured that produced very high quality castings. Purchased parts were of good quality and were functionally acceptable. Tooling and fixturing were fabricated in-house for machining, balancing, and assembly. Outside sources were developed for those tasks requiring very specialized equipment or skills. Figures 2 through 11 are photographs of the resulting parts.

There are 24 items, consisting of piece parts and sub assemblies, which make up the turbocharger. They are shown in Figure 12.

FIGURE 2 - COMPRESSOR HOUSING

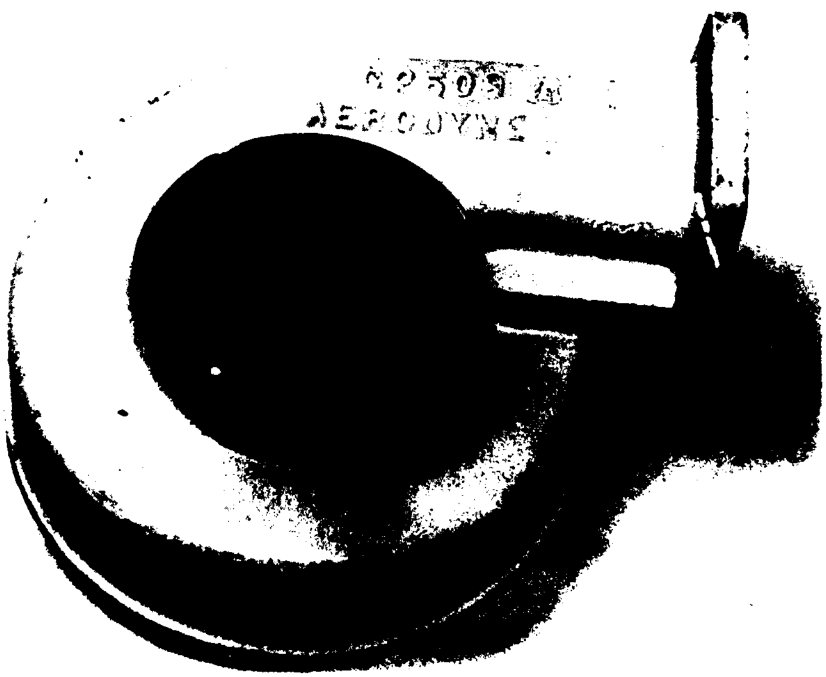


FIGURE 3 - TURBINE HOUSING

FIGURE 4 - COMPRESSOR BACKWALL

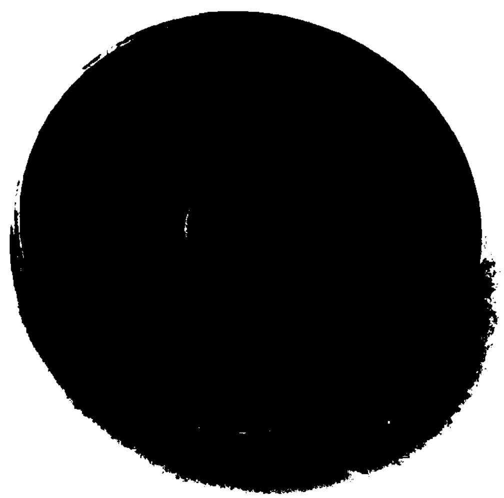


FIGURE 5 - TURBINE BACKWALL

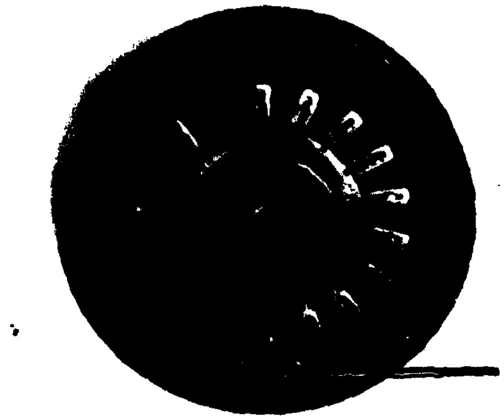


FIGURE 6 - TURBINE BACKWALL WITH HEAT SHIELD AND CONTROL LEVERS

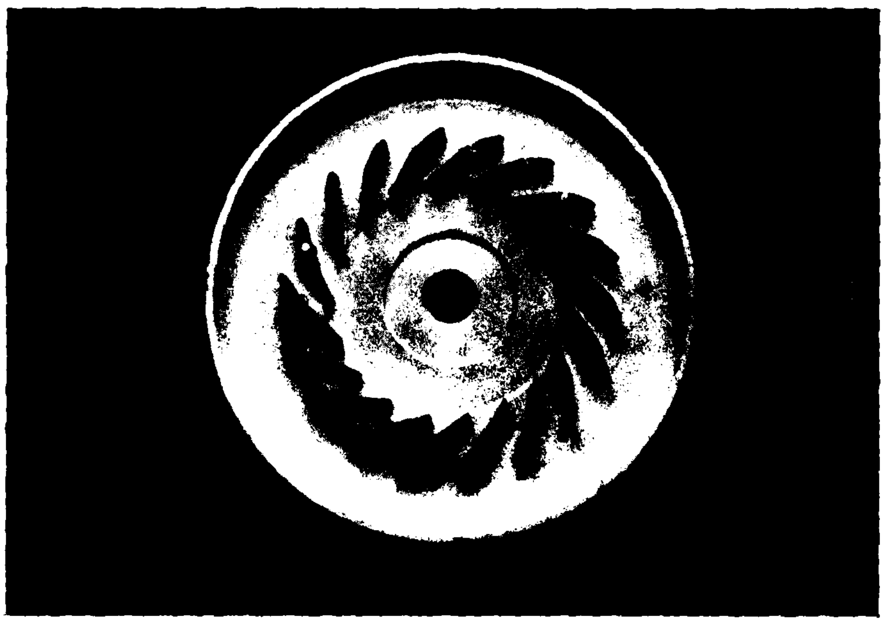


FIGURE 7 - TURBINE BACKWALL AND NOZZLES

FIGURE 8 - COMPRESSOR WHEEL CASTING

FIGURE 9 - TURBINE WHEEL CASTING

FIGURE 10 - TURBOCHARGER CLAMPS

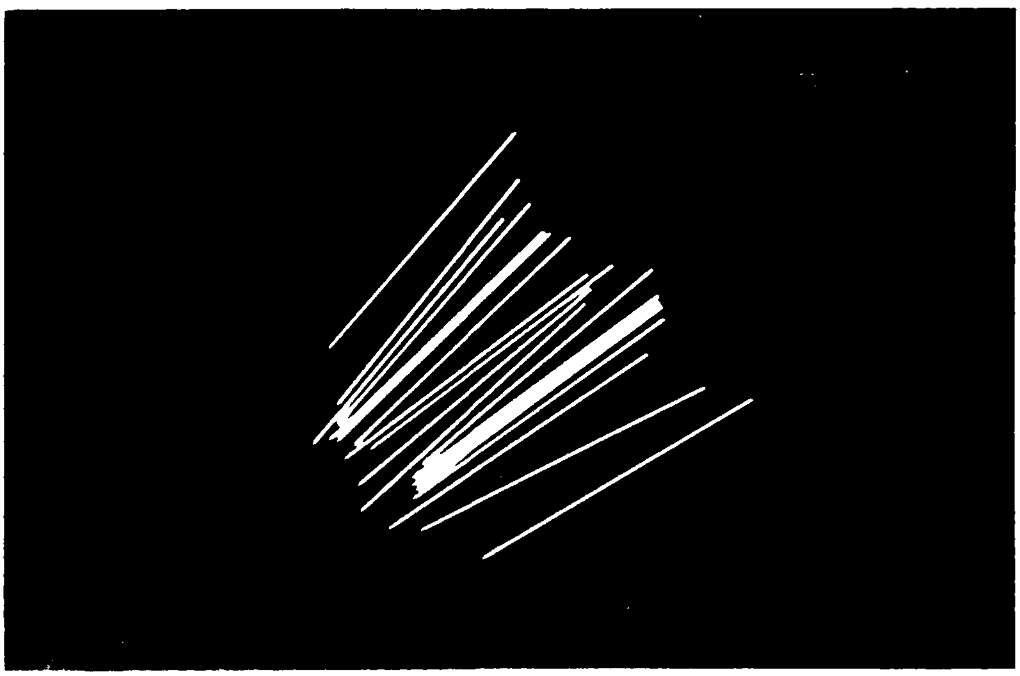


FIGURE 11 - OIL WICKS

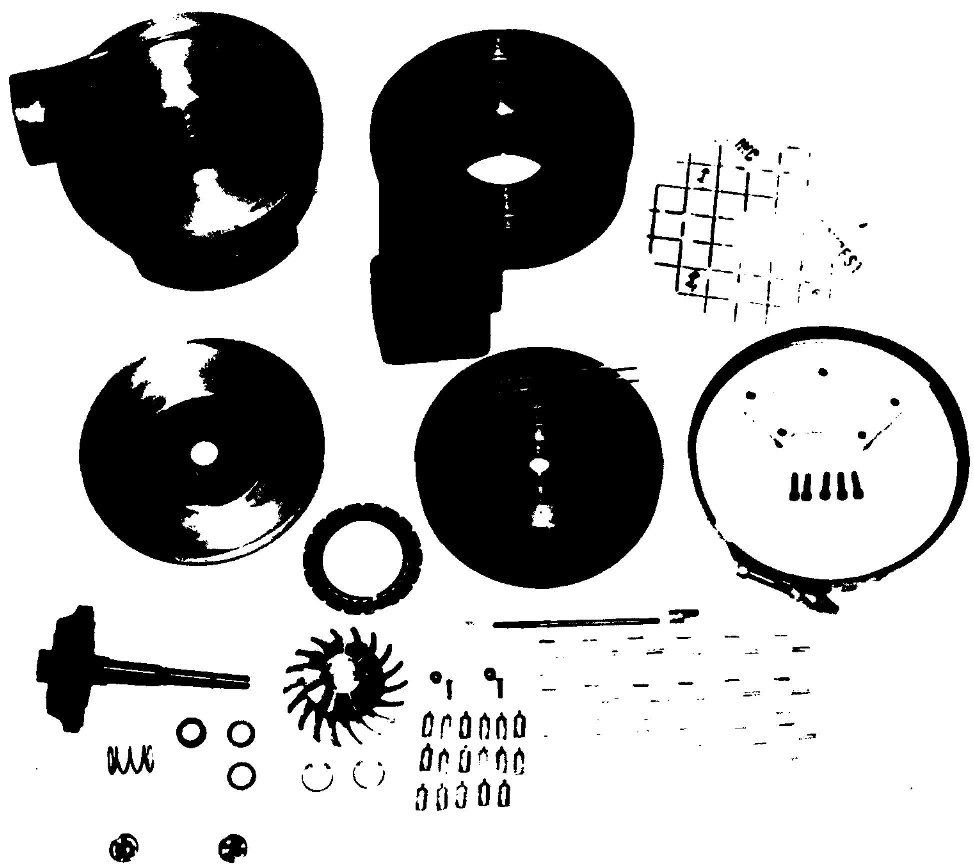


FIGURE 12 - TURBOCHARGER PIECE PARTS AND SUBASSEMBLIES

B. Bench Testing

The bench testing consisted of conducting oil/wick/wick-shaft interface tests and running of complete turbochargers.

1. Oil/wick/wick-shaft interface tests

The purpose of these tests was to determine the oil flow rate of the lubrication system (comparing results with the result found on the rotor rig test ran earlier) and to evaluate the flow characteristics of two different candidate oils as well as the selected wick material.

The flow rate found after 231.5 hours of single rotor rig test was .0069 cubic inches of oil (Mobil DTE medium) per hour for the two wicks. In this case the shaft was run vertically with no opportunity for recirculation of the oil.

For the present test a test rig, simulating the slinger ramps on the turbocharger shaft, was utilized. The surface speed of the slingers represented a rotational speed of 130,000 RPM of the turbocharger rotor. Twelve ramps were incorporated on the test rig rotor. Provision for 12 wicks and 12 graduated cylinders was made. A photograph of this rig is shown on Figure 13.

For these tests the original spindle oil (Mobil DTE medium) and a turbine oil (Humble Turbo Oil #2380-MIL-L-23699B) were used.

Two test conditions were run - the first allowed the oil that was ejected from the ramp to collect around the wick and therefore had an opportunity to recirculate. The second condition shielded and drained the wick so there was no opportunity for recirculation of the oil.

The average of the results are as follows (flow for two wicks):

RECIRCULATION ALLOWED	NO RECIRCULATION ALLOWED
Mobil DTE medium	Mobil DTE medium
.00135 in^3/hour	.00739 in^3/hour
Humble Turbo Oil # 2380	Humble Turbo Oil # 2380
.00128 in^3/hour	.00669 in^3/hour

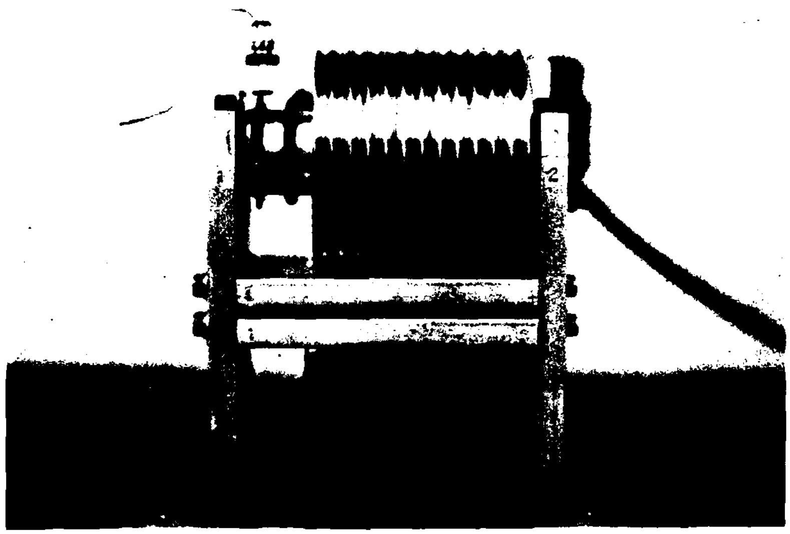


FIGURE 13 - OIL WICK TEST RIG

After preliminary running of the rig to establish pulley ratios, rotor speed and overall operating characteristics, the wicks, (first soaked in the appropriate oil) were placed in the rig and adjusted for the appropriate contact. The graduated cylinders were filled with the selected oil and mounted in the rig. Readings were taken on all cylinders and the test was run continuously for 138.5 hours. Following this test five wicks were replaced and the rig set up to eliminate the possibility of recirculation. This test was run continuously for 136 hours.

2. Complete turbocharger testing

The purpose of this testing was twofold:

* Mechanical - Determine the basic integrity of the turbocharger components and develop the bearing system to the point that the engine testing could be attempted with some degree of confidence.
* Aerodynamic - Generate compressor maps and verify turbine performance and its ability to control power output through the VATN.

The results of turbocharger testing were:

* Mechanical - The basic integrity of the turbocharger components, as designed, was shown to be adequate. There were no failures of component due to steady or vibratory stresses (other than bearing failures). The rotor has been run (cold) to a speed of 205,000 RPM. Lubrication of the bearings proved to be adequate. The bearing geometry had to be accurate and balance requirements were very important (as expected).
* Aerodynamic - Data was obtained to construct complete compressor maps of the "as designed" compressor as well as "high flow" and "low flow" trims of the basic compressor. These compressors utilized a vaned diffuser. Additionally, a vaneless version of the "high flow" compressor was tested and a compressor map constructed. These maps are shown on Figures 14, 15, 16, and 17.

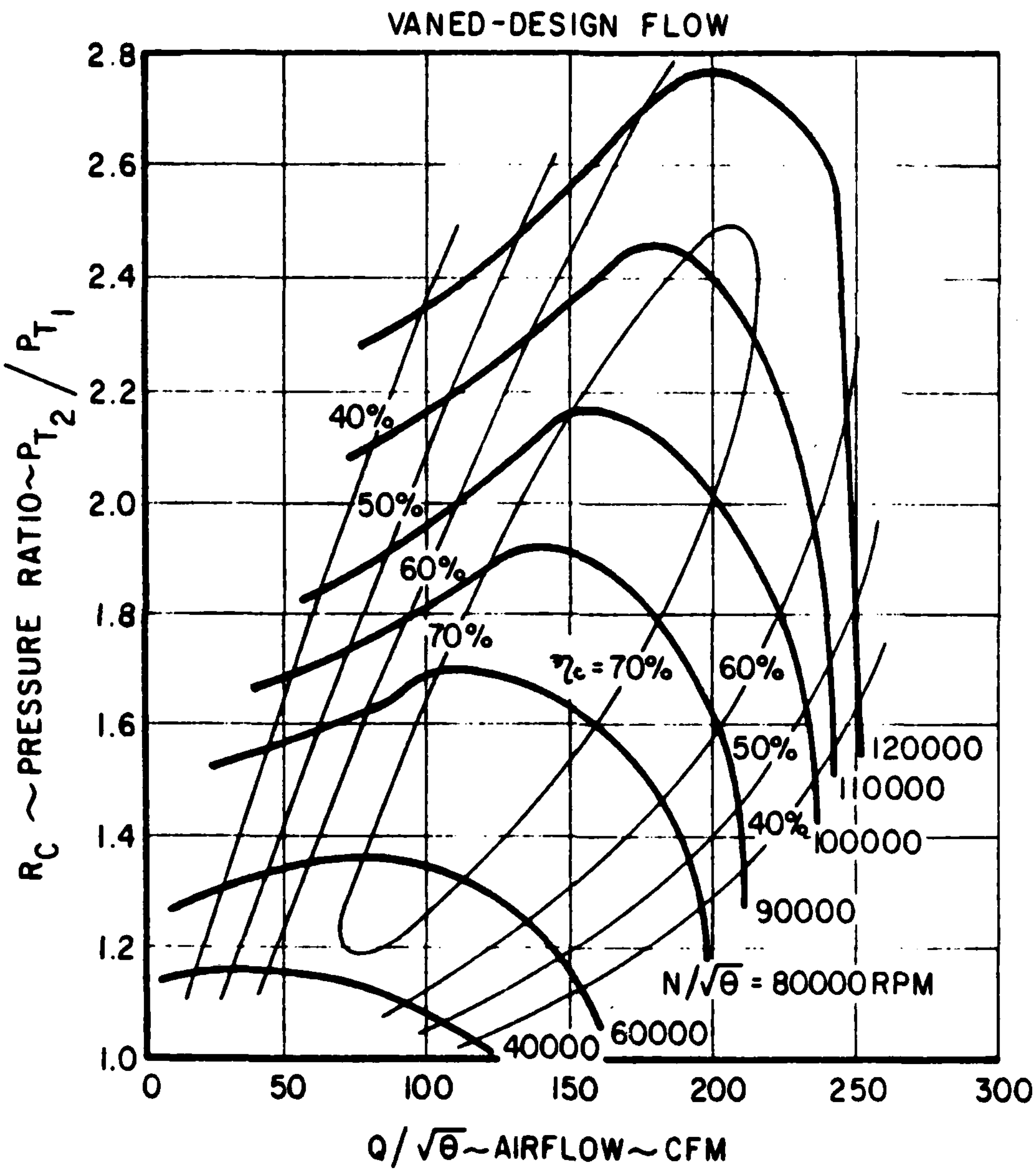

VANED-DESIGN FLOW
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0
50
100
150
200
250
300
R_C ~ PRESSURE RATIO ~ P_{T_2}/P_{T_1}
$Q/\sqrt{\theta}$ ~ AIRFLOW ~ CFM
40%
50%
60%
70%
η_c = 70%
60%
50%
40%
120000
110000
100000
90000
$N/\sqrt{\theta}$ = 80000 RPM
60000
40000

FIGURE 14

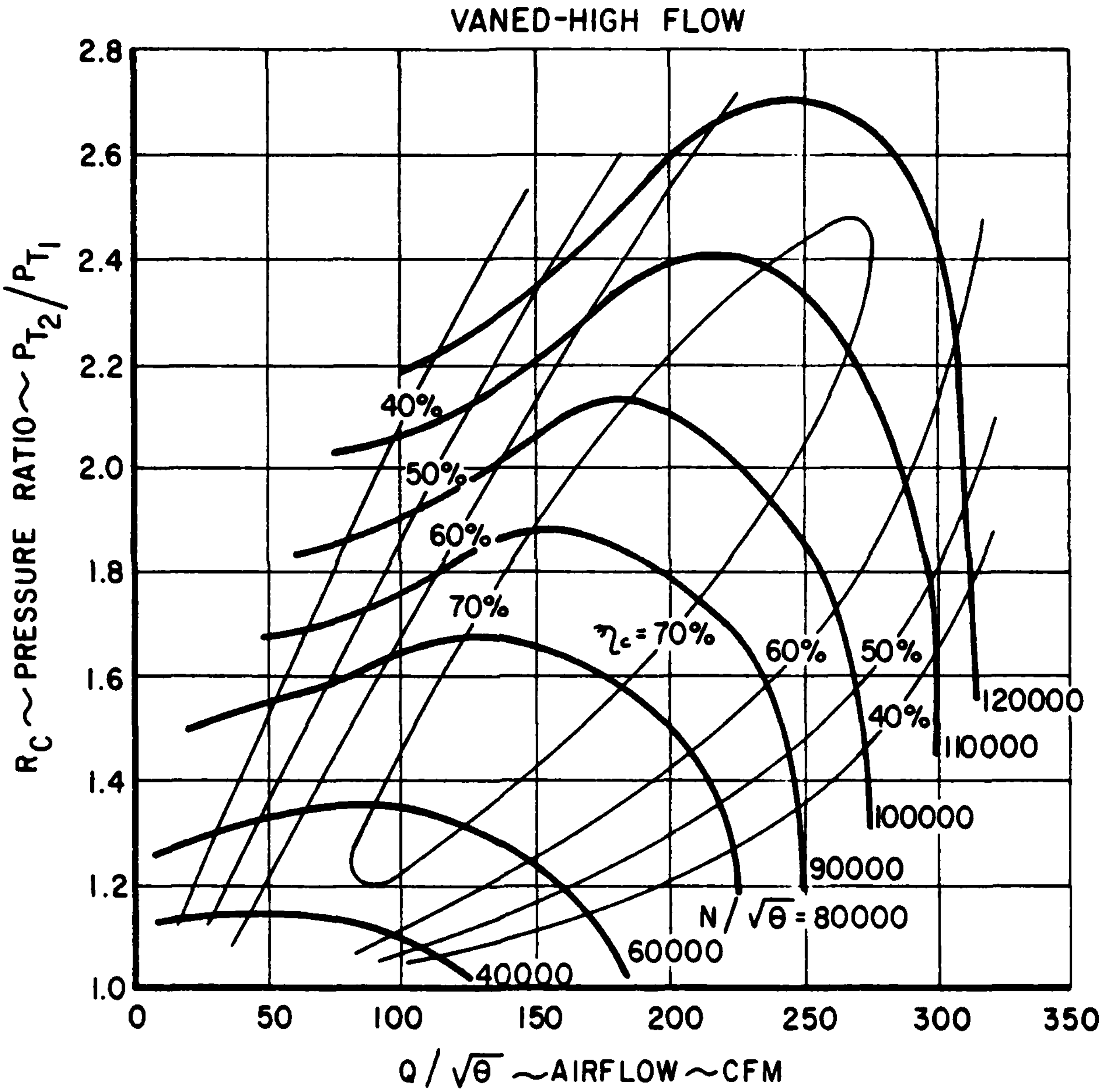

VANED-HIGH FLOW
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0
50
100
150
200
250
300
350
40%
50%
60%
70%
η_c = 70%
60%
50%
40%
120000
110000
100000
90000
N / √θ = 80000
60000
40000
R_C ~ PRESSURE RATIO ~ P_{T_2}/P_{T_1}
Q / √θ ~ AIRFLOW ~ CFM

FIGURE 15

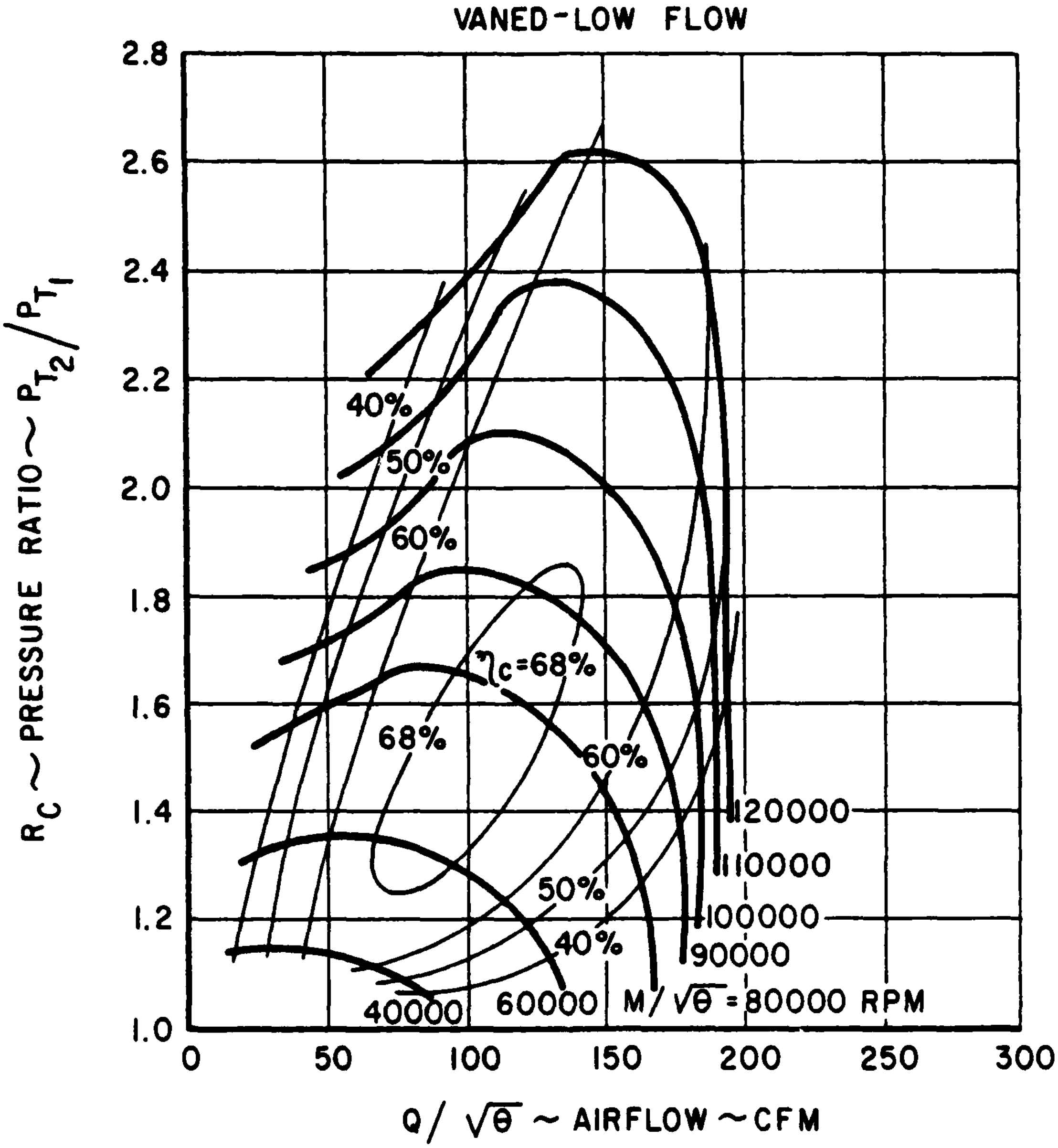

VANED-LOW FLOW
2.8
2.6
2.4
2.2
2.0
1.8
1.6
1.4
1.2
1.0
0
50
100
150
200
250
300
$R_C \sim$ PRESSURE RATIO $\sim P_{T_2}/P_{T_1}$
$Q/\sqrt{\theta} \sim$ AIRFLOW $\sim$ CFM
40%
50%
60%
η_c=68%
68%
60%
50%
40%
120000
110000
100000
90000
40000
60000
$M/\sqrt{\theta}$ = 80000 RPM

FIGURE 16

VANELESS-HIGH FLOW

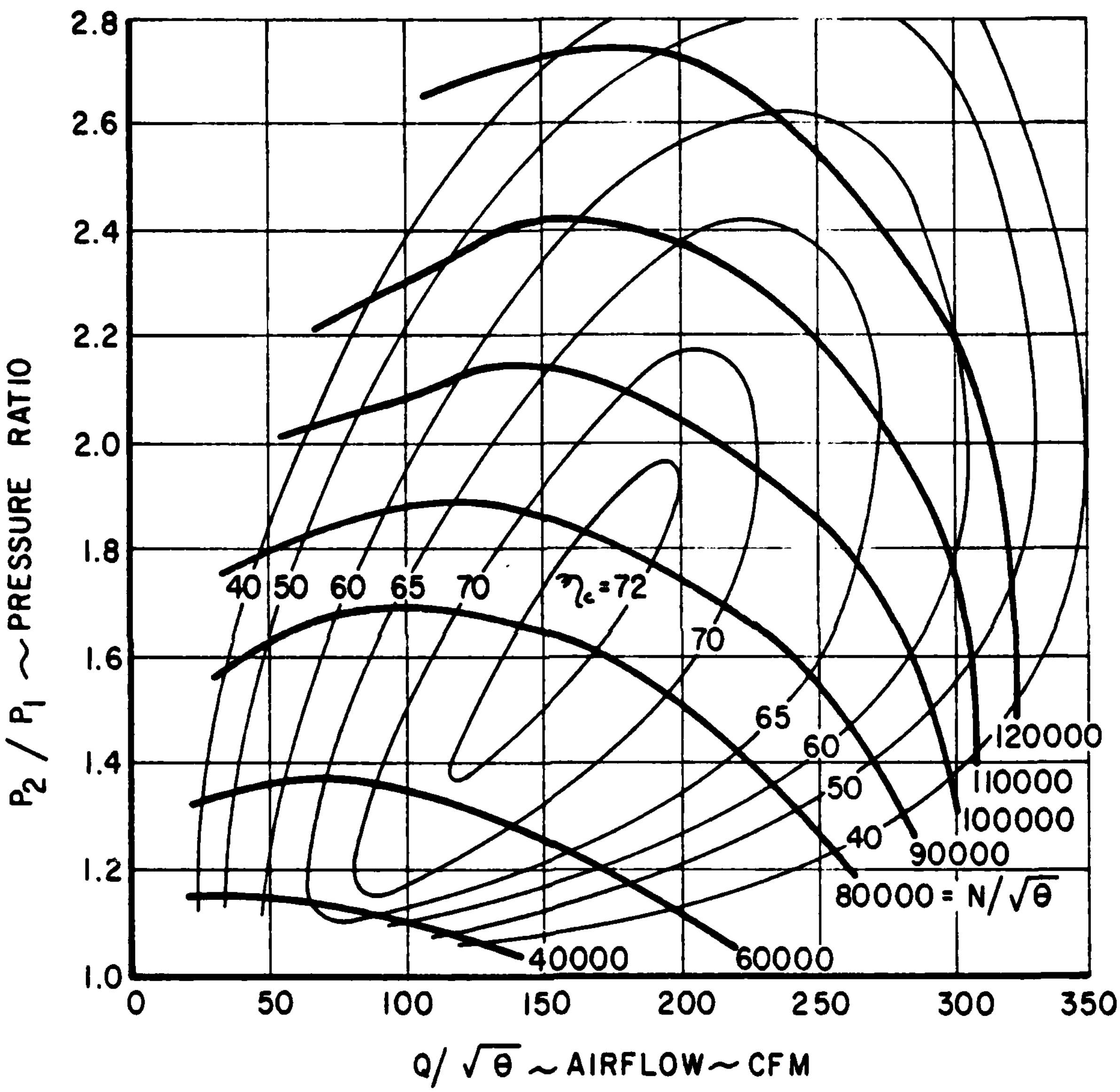


FIGURE 17

Turbine testing was accomplished to the point of verifying design goal efficiencies and demonstrating the ability of the VATN to control power output and therefore rotor speed.

In order to conduct the mechanical and aerodynamic tests a complete facility, including a data acquisition system, had to be designed and built. An outline of the features of this facility are presented in Appendix C.

C. Develop Mathematical Models

The objective of this effort was to develop mathematical modeling techniques whereby the effects of turbocharging on diesel engine characteristics could be predicted - both from a theoretical standpoint and using actual test data from an empirical standpoint. A further objective was to utilize these predicted characteristics to evaluate the effects of engine displacement and drive ratios on fuel economy and emissions for a typical duty cycle.

A computerized mathematical model was developed to theoretically compute the fuel used over the 13 Mode Federal Diesel Emission Cycle for diesel engines. Another computerized mathematical model was developed to calulate the fuel used over the Federal Urban and Highway Driving Cycles using empirical relations developed from the turbocharged engine test data. The model is based on empirical formulae derived from experimental data of various engines by C. F. Taylor (Reference 1 of Appendix D). The duty cycle is divided into many short "steady state" conditions and the fuel consumed at each condition calculated. Total fuel consumption is the summation of all conditions.

This work was conducted by Southwest Research Institute and the details of the effort are included in their report which is attached as Appendix D.

D. Select Engine

The objective of this effort was to select a commercially available four stroke diesel engine that would have a swept volume rate (displacement X RPM) that would closely approximate the pressure flow characteristics of the proposed turbocharger compressor. Secondary considerations such as availability, being previously turbocharged, etc. were included.

A John Deere, 4 cylinder, direct injected, turbocharged engine was selected. It's displacement is 239 cubic inches and maximum speed is 2500 RPM.

This work was conducted, in large part, by Southwest Research Institute and the details of the effort are included in their report which is attached as Appendix D.

E. Engine Performance Tests

The objective here was to conduct engine performance tests, collecting fuel consumption, CO, NO_2, HC and smoke emissions data for the baseline turbocharged engine, the engine with no turbocharger and with the Aerodyne turbocharger at various VATN settings. Additionally, transient response characteristics were to be evaluated.

After a "break-in" period a matrix (speed-load) of engine data was obtained for the turbocharged engine "as-received" and without the turbocharger. Then, with the Aerodyne turbocharger installed, the same speed-load matrix testing was accomplished for three different turbine nozzle settings at each speed-load point. Transient tests were conducted. A total of 53 hours of testing was accomplished with the Aerodyne turbocharger. Figures 18, 19, and 20 graphically summarize the results of the performance testing.

This work was conducted by Southwest Research Institute and all of the reduced test data and the details of the effort are included in their report which is attached as Appendix D.

F. <u>Predict Fuel Consumption</u>

The objective was to show the potential improvements available through turbocharging with an advanced technology turbocharger in a typical automotive duty cycle. Two

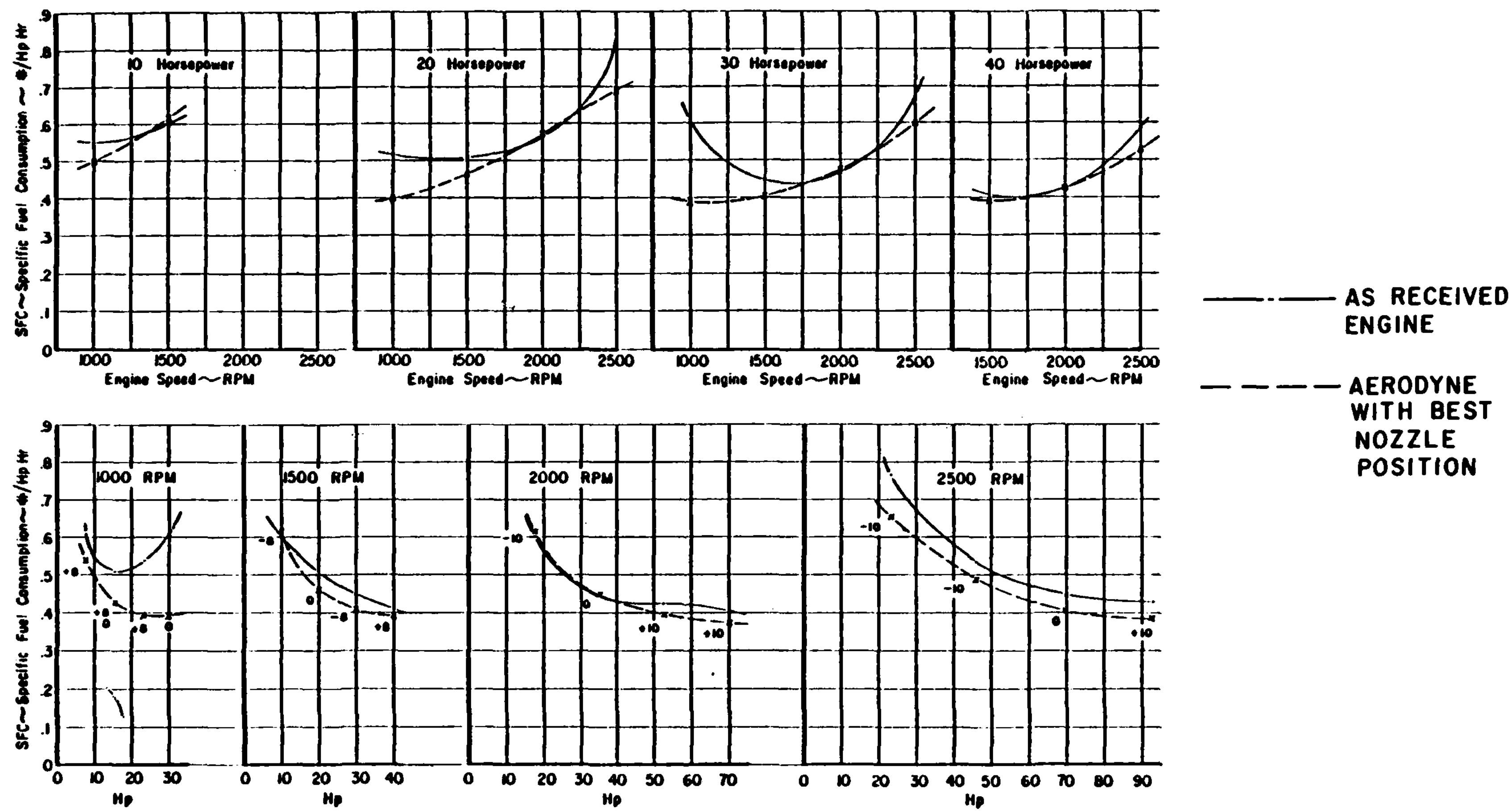
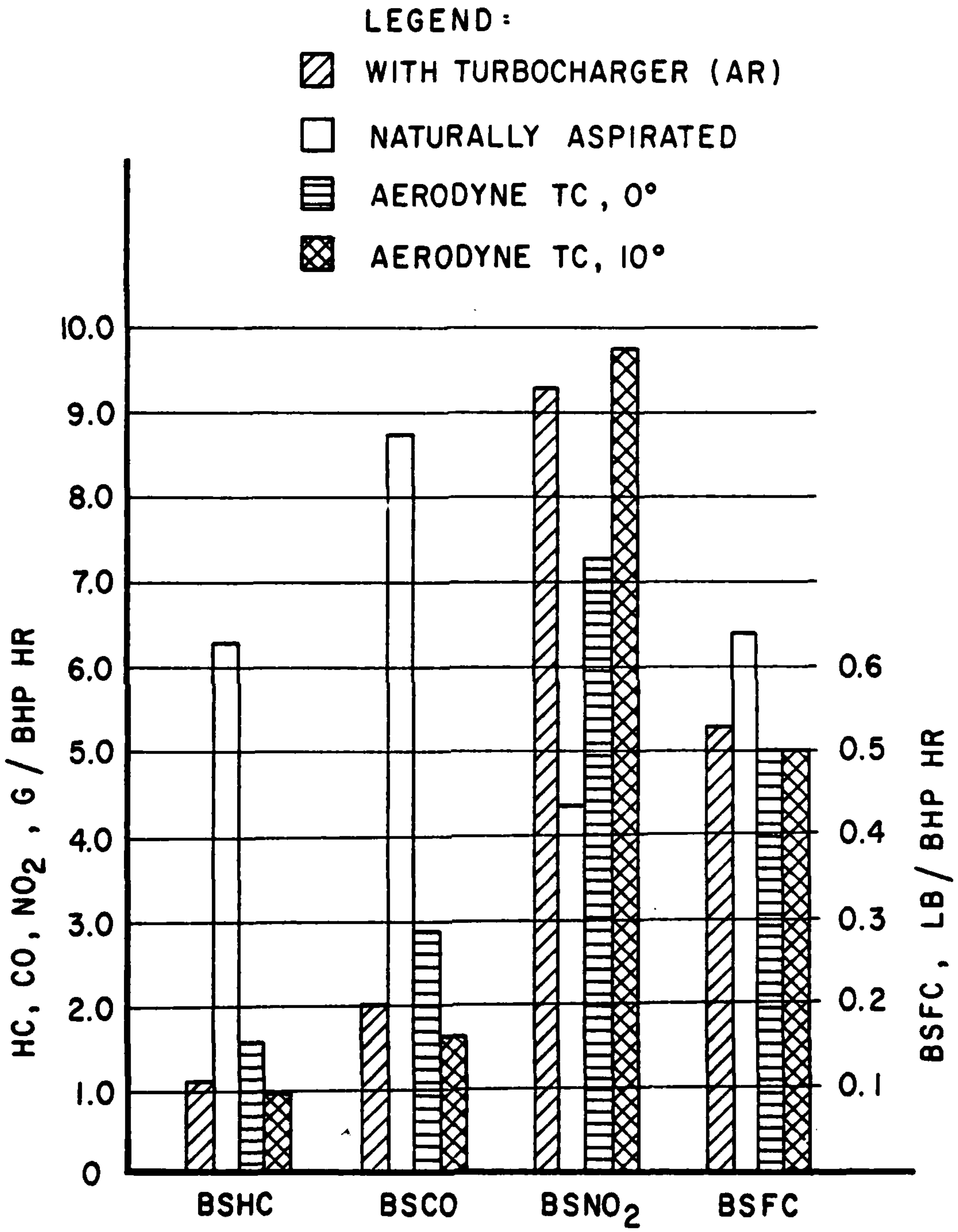
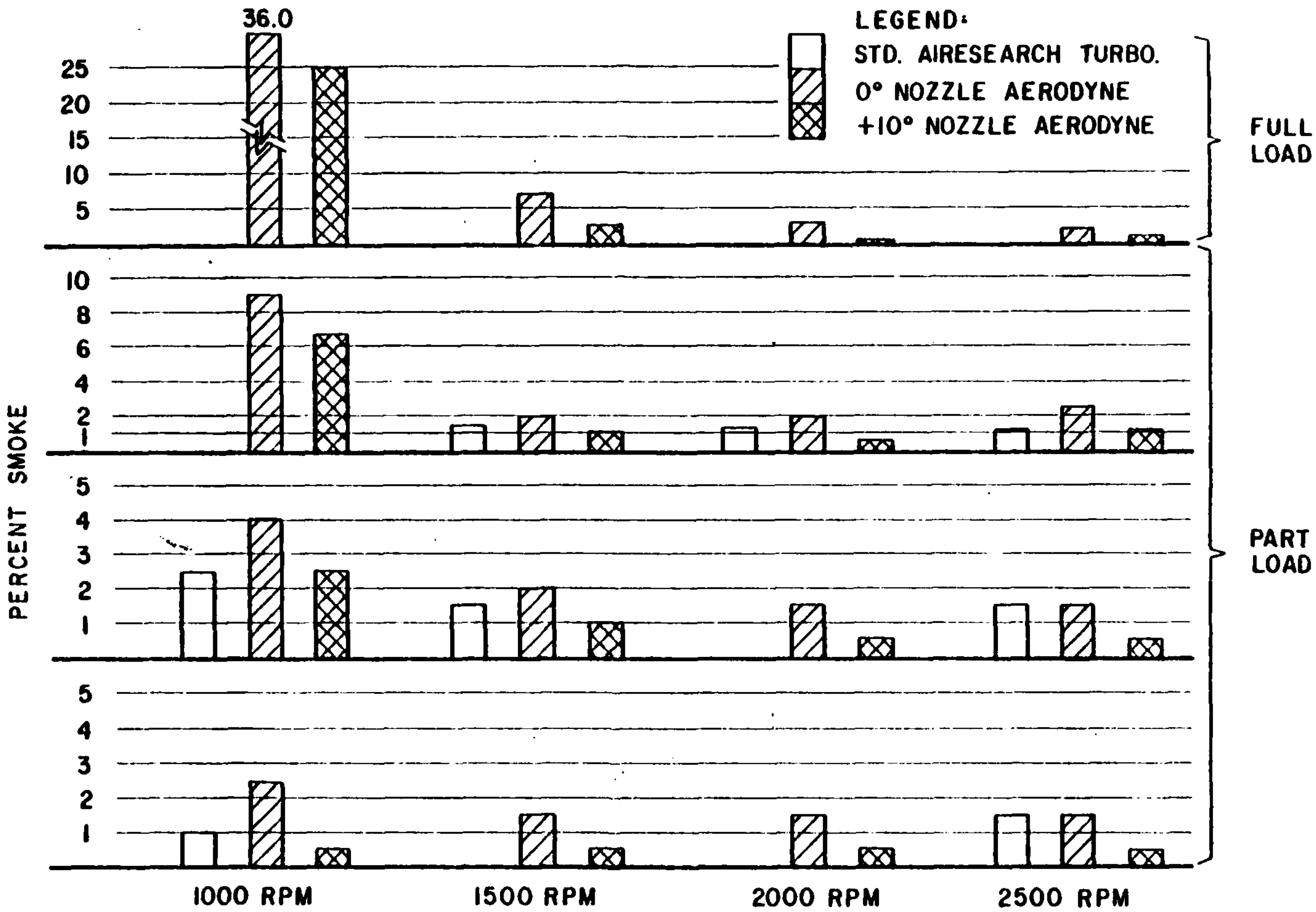


FIGURE 18 - SPECIFIC FUEL CONSUMPTION Vs RPM & BHP



EMISSIONS AND FUEL ECONOMY OVER 13-MODE FEDERAL DIESEL EMISSION CYCLE

FIGURE 19



BAR PLOTS OF SMOKE TEST RESULTS

FIGURE 20

primary sources of improvement would be utilized.

(1) At any given engine operating point (load/ speed), optimize the VATN setting to produce minimum fuel consumption.

(2) Through final drive ratio changes, cause the engine to operate at various BMEP levels. This will cause variations in internal engine friction as well as basic overall thermodynamic efficiency. As the engine is forced to run slower and slower, performance would be made up through higher levels of turbocharging.

This effort was conducted via the previously developed computerized mathematical model for the Federal Urban and Highway Driving Cycle using empirical relations derived from the engine test data. The results of this analysis are shown in Figure 21, which is a plot of fuel consumption versus final drive ratio (expressed as engine speed/vehicle velocity) at optimum turbine nozzle position.

This work was conducted by Southwest Research Institute. The results of the analysis and the details of the effort are included in their report which is attached as Appendix D.

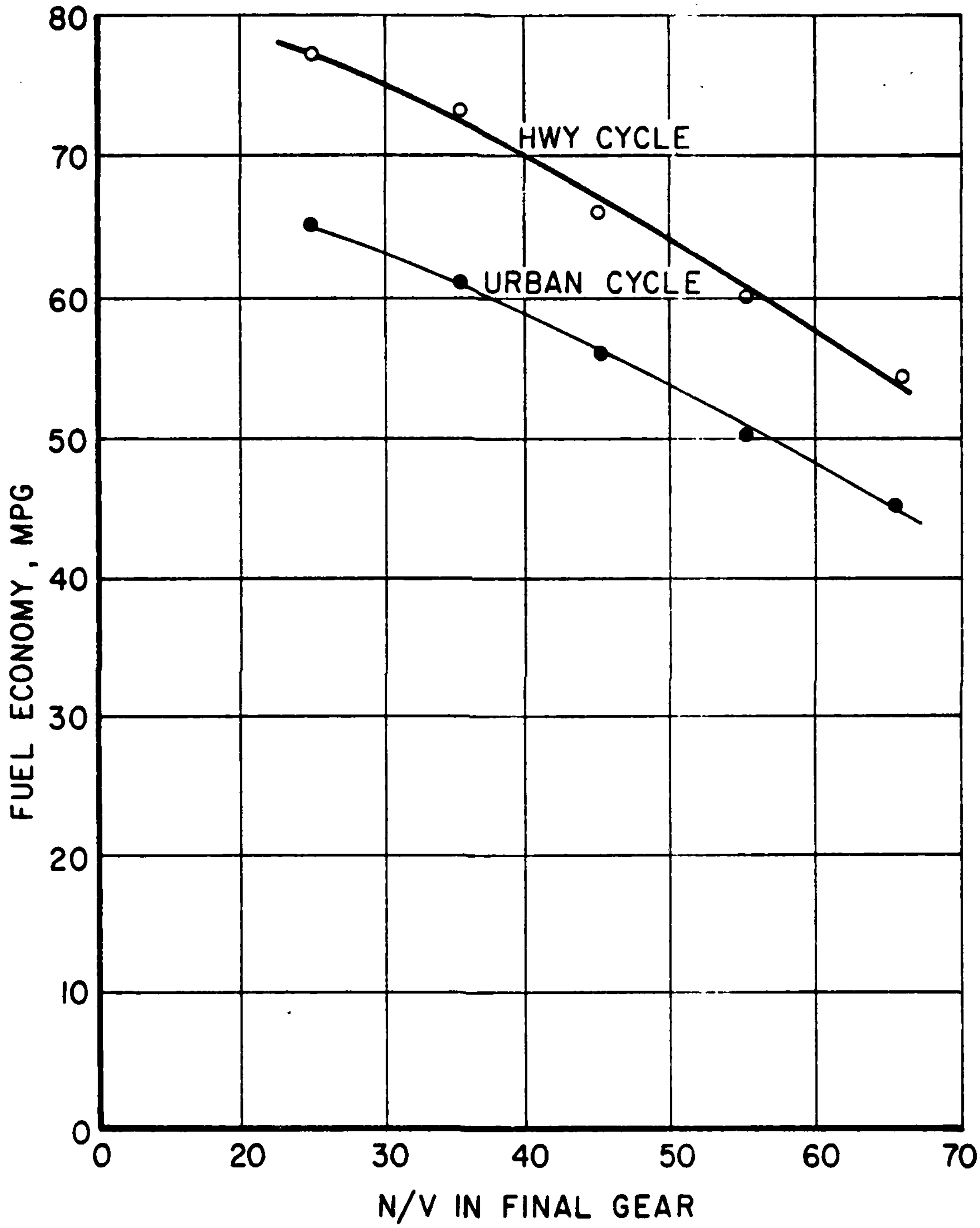


FIGURE 21-MODEL ESTIMATION OF FUEL ECONOMY FOR VARIOUS N/V RATIOS

V. DISCUSSION

A. Manufacture Turbochargers

The assembly of the Aerodyne turbocharger is simple and quick (particularly compared to normal turbochargers). While the VATN adds to the complexity and assembly time, this is outweighed by the simplicity of the bearing system and lack of a bearing housing, thrust plates and washers, piston rings and "O" rings. In-house studies indicate that little or no cost difference exists between this design and a typical wastegated turbocharger.

The Aerodyne turbocharger weighs about 10.5 pounds compared to 16-17 pounds for similar flow size commercially available turbochargers. At this point, all parts in the turbocharger can be produced on a prototype basis. The turbocharger was designed with producibility as a keystone design objective. All indications are that all parts can be mass produced readily.

B. Bench Tests

The wick testing supported the basic lubrication system design approach. Adequate consistency was demonstrated. Allowing the oil to recirculate reduced the consumption by a factor of perhaps five and may prove to be a means of minimizing oil consumption.

Mechanically, the turbocharger proved to be sound. Quite often turbomachinery is beset with vibratory stress problems leading to fatigue failures. To date no such problems have been found. The demonstration of 205,000 RPM (about 80 percent overspeed) produced a permanent set of about .004 inch in the compressor wheel but showed the basic integrity of the rotating components. The turbocharger was once operated for about three hours at 110,000 RPM at the lowest attainable flow (about 32 CFM) with no detrimental effects.

The compressor maps produced with the basic compressor hardware are unique. While backward curved blading is known to produce a less pronounced surge, the familiar compressor map surge line still exists and operation to the left of this line is not practical. The characteristics of this compressor hardware are such that the surge line is very difficult to define and, more importantly, operation to its' left, on the map, produces no ill effects. The air produced in this region of the map is very usable by an engine (as practically demonstrated on the John Deere engine). What this allows is the turbocharging of an engine at any speed so long as the turbine can produce the required power.

The compressor efficiencies demonstrated in these tests (for the "as designed" and for flow trim modified compressors) are as good or better than published data for similar flow compressors. The design flow compressor achieved a peak efficiency of 76 percent. The vaneless configuration achieved about equal peak efficiency with published data but showed a much broader operating range.

As a means of providing comparitive data, a compressor map was constructed for the turbocharger received with the test engine. The data was obtained on Aerodyne's test rig using the same instrumentation as on all other tests. This compressor map is shown on Figure 22.

Sufficient turbine performance data was obtained to show that the design goal peak efficiency of 75 percent was met and that the VATN produced radical changes in power output while maintaining good efficiency. Evaluation of the turbocharger, with VATN, on an engine, from bench test data was beyond the scope of this effort.

"First order" estimates of turbine performance were made from the data taken at both Southwest Research Institute and Aerodyne. The results of these estimates are shown

COMPRESSOR MAP - TURBOCHARGER
RECEIVED ON JOHN DEERE ENGINE

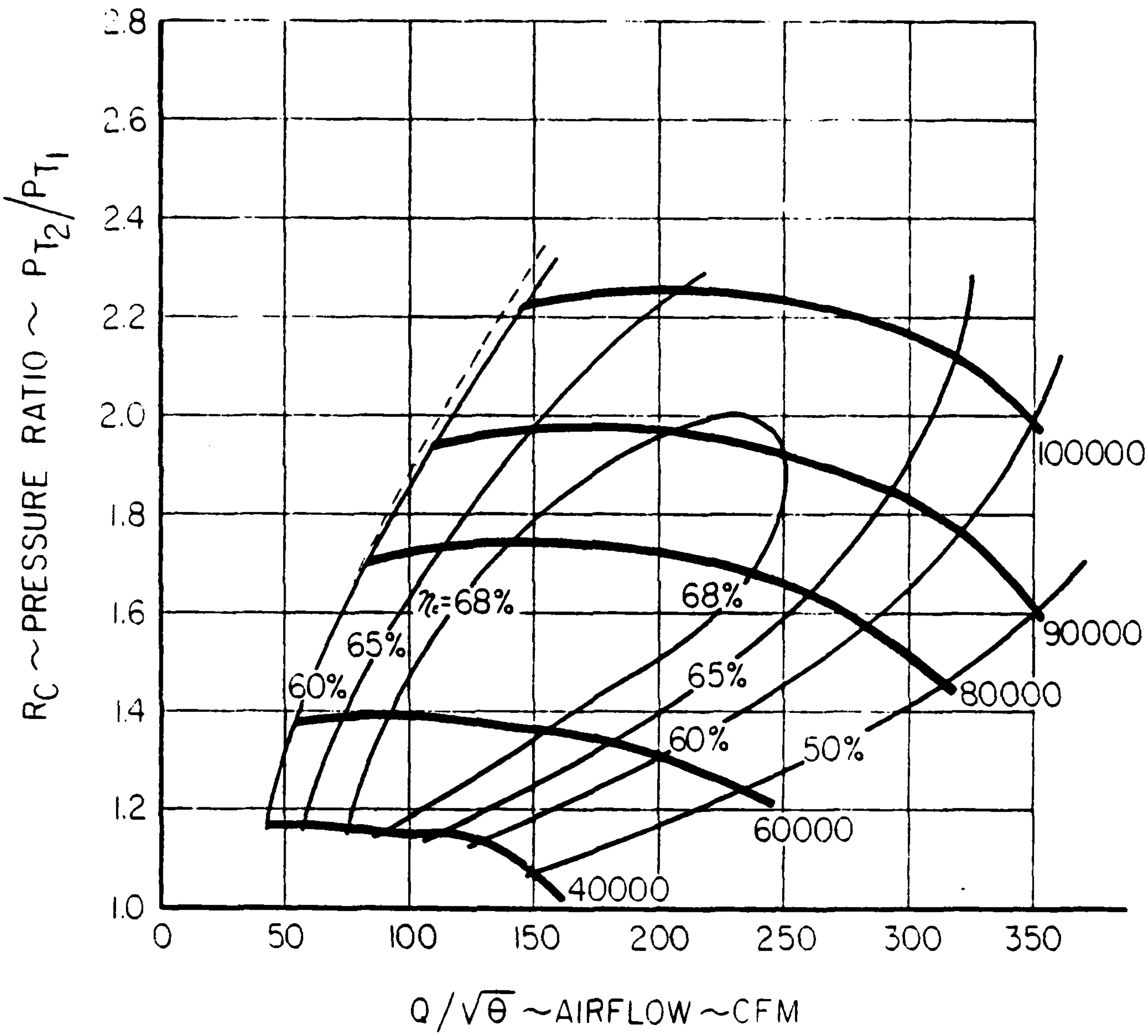


FIGURE 22

on Figure 23. The SWRI results were calculated as follows:

Using measured compressor pressure ratio and corrected airflow in conjunction with the compressor efficiency from the map generated at Aerodyne for this compressor, the required work was calculated. Measured turbine inlet temperature, turbine inlet pressure and turbine exit static pressure were then used to calculate ideal turbine work, corrected turbine flow and V' (isentropic jet velocity). Turbocharger speed was deduced from the compressor map since many of the speed readings seemed suspect.

A similar method was used to calculate turbine efficiency from the results of three different compressor component tests. Figure 23 shows that the Aerodyne data was all at conditions greater than the optimum U/V' (U=turbine rotor tip speed) and all the SWRI data was taken at U/V' values less than optimum (classically, this curve normally shows a peak efficiency in the range of .65 to .70 values for U/V'). Therefore, the maximum efficiency of the turbine was not observed. It is felt that, from the data plotted in Figure 23, the maximum total-to-static efficiency might be in the 77 to 78 percent range. This would reflect

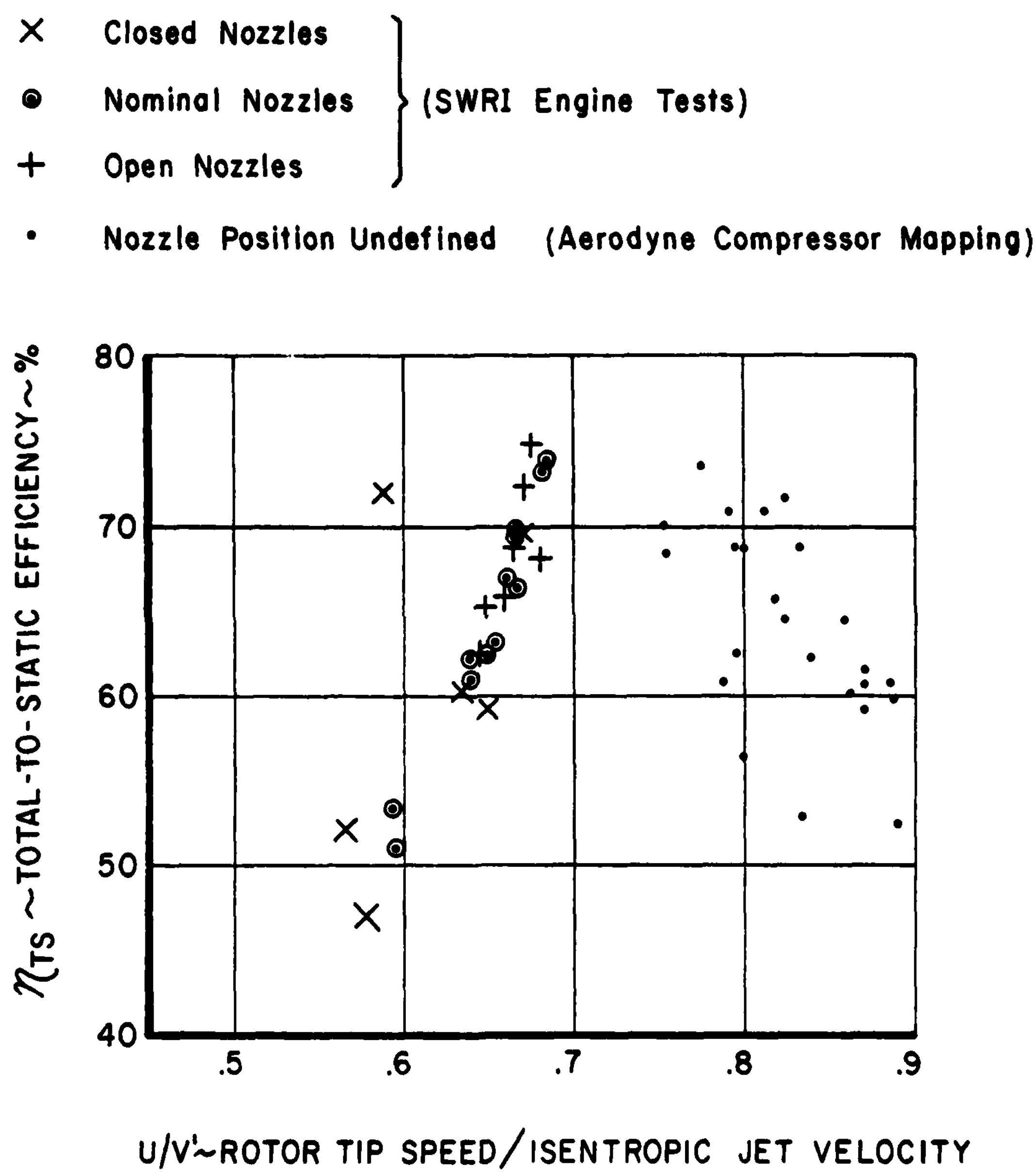


FIGURE 23 - MEASURED TURBINE EFFICIENCY

peak total-to-total efficiency of about 80 percent.

C. Engine Performance Tests

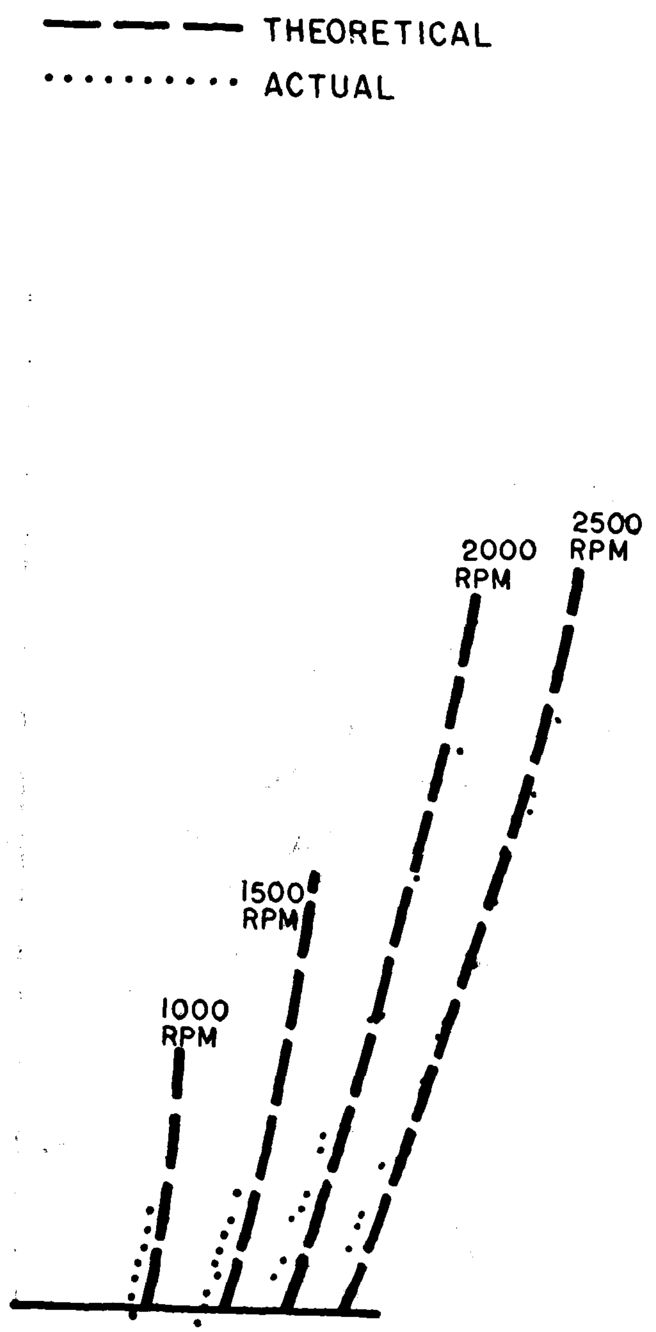
The turbocharger design was complete before the contract was begun and the engine was selected to match the anticipated flow characteristics. The aerodynamic design point was: a corrected compressor flow of 200 CFM at a compressor pressure ratio of 2.3 and a turbine inlet temperature of 2060°R at a fuel/air ratio of .067 with compressor inlet loss of 1 inch of mercury and a turbine discharge loss of 6 inches of mercury.

The engine testing produced compressor pressure ratios that were generally much lower than design point and the maximum turbine inlet temperature encountered was only 1650°R. Secondary differences were that the design point losses were not reached in the engine testing. Therefore, while complete turbine maps are not available, it must be assumed that the turbine was operating far from peak efficiency. Review of the actual engine pressure/flow characteristics plotted on the actual compressor map (see Figure 24) reveal that the compressor flow potential was greater than optimum. The maximum engine speed line (2500RPM) should have been further to the right such that the 2000 RPM line was in the peak efficiency area of the compressor map.

Also, since the turbine data did not show a peak in turbine efficiency (see earlier discussion of turbine efficiency versus U/V'), it can be concluded that the turbine and compressor aerodynamic matching can be improved thereby producing higher turbine efficiencies.

The back-to-back compressor/turbine arrangement, of necessity, brings the turbine and compressor flowpaths in close proximity to each other, thereby producing the opportunity for transfer of heat from the turbine to the compressor. For turbocharging a diesel engine this is an adverse condition since the added heat in the compressor flow results in a density decrease (opposite the desired result). The turbocharger, as designed, incorporates an air gap and an asbestos heat shield as a barrier to heat flow. However, there is a rather direct metalic path for heat flow at the outer diameter of the turbocharger where the piloting and clamping of the compressor and turbine stationary parts takes place. Compressor discharge temperature data on the test engine did not match the theoretical temperature that would be calculated from the compressor map - showing that heat transfer was taking place. This heat transfer was of such a magnitude that a significantly adverse effect on airflow resulted (see Figure 24). An analysis of the engine performance penalties resulting from the heat transfer is beyond the scope of this effort. A number of ways exist to modify the heat transfer characteristics, including; slots, additional barriers and material changes.

The "as designed turbocharger was capable of plus or minus 11 degrees of turbine nozzle vane travel. Some of the initial testing was conducted at plus and minus 8 degrees as well as nominal. Later testing was at plus and minus 10 degrees (and nominal). The engine test data reveal that the potential improvements (or penalties for that matter) had not yet been reached, under most operating conditions, at even the 10 degree extreme of motion. In other words, additional turbine nozzle travel would have produced an additional incremental decrease (or increase - according to operating condition and direction of movement) in specific fuel consumption and emissions.



THEORETICAL AND ACTUAL AIRFLOW
PLOTTED ON COMPRESSOR MAP

FIGURE 24

Taking the above into account (poor compressor and turbine match, excessive heat transfer and the potential for utilizing even more turbine nozzle travel) and the fact that this was not a fully developed turbocharger - the specific fuel consumption results were very encouraging. The specific fuel consumption was improved over nearly the entire load/speed range of the engine (except medium speed/medium to high load where fuel consumption was about equaled) compared with the "as received" engine. Emissions, which are primarily sensitive to fuel/air ratio at a given speed/load condition, could be made better or worse through nozzle position changes. It is estimated that an additional four percent improvement in sfc can be achieved through a rematch of compressor and turbine. Reducing the heat transfer and providing additional nozzle travel should produce an additional two to three percent improvement at the extreme operating conditions.

D. Predict Fuel Consumption

The analysis shows clearly that very significant improvements in fuel consumption are available through turbocharging - particularly via drive ratio changes (causing the engine to run slower and at higher BMEP). Performance is made up through turbocharging. A strong secondary influence that is available using an advanced turbocharger, such as the present one, is through the optimization of fuel/air ratio, manifold pressures and the like by means of the VATN. Unfortunately, the model did not predict the degree of turbocharging required to maintain performance as drive ratios were changed. However an advanced turbocharger can more readily achieve this goal because of its ability to produce boost at very low engine speeds. Additionally, in light of the discussion in the previous section, adequate opportunity exists for even further improvements in fuel consumption via turbocharger performance improvements.

VI. CONCLUSIONS

A. Manufacture Turbochargers

1. Turbochargers of this design were built using present day manufacturing techniques and were successfully operated.
2. The manufacturing techniques used lend themselves to mass production techniques and studies show that cost would be comparable to present technology turbochargers.
3. The Aerodyne turbocharger is inherently lighter in weight than present technology turbochargers owing largely to the lack of a separate bearing housing.

B. Bench Tests

1. The bearing system allows stable rotor operation throughout the operating range of the turbocharger. Compressor blade tip clearance of .005 inch and turbine blade tip clearance of .008 can be run with no interference.
2. The lubrication system provides adequate lubrication for the bearings with enough oil in the present turbocharger reservoir for at least 680 hours of high speed operation.
3. Compressor efficiencies equal or exceed present equivilent flow turbocharger compressors, but, with a broader operating range. No defined surge

exists and operation to the left of stall is practical.

4. Turbine efficiencies met design goals.

5. The VATN produced large changes in power output, but effects could not be evaluated at this stage of testing.

6. There are no inherent detrimental vibration or stress problems with any component or assembly of the turbocharger.

C. Mathematical Models

1. The present model gives a good representation of the relative effects on fuel consumption of turbocharging and final drive ratio changes.

2. The model does not address emissions.

3. The model does not predict maximum turbocharging levels required to meet minimum performance requirements.

D. Engine Performance Tests

1. An advanced turbocharger with VATN can produce significant improvements in specific fuel consumption including a "flattening" effect on specific fuel consumption versus load at a constant engine speed or specific fuel consumption versus speed for a

constant power level.

2. The aerodynamic match of both the compressor and turbine of the present turbocharger was poor.

3. The heat transfer from turbine to compressor was excessive.

4. Plus and minus 10 degrees of VATN vane excursions was not adequate to show the extremes of potential improvement.

5. A VATN system can significantly affect emissions via A/F ratio control.

6. An extremely broad range of pressures and air flows can be run with this type turbocharger.

E. <u>Predict Fuel Consumption</u>

1. Turbocharging can be used to effect dramatic improvements in fuel economy for a diesel engine in an automotive application.

2. An advanced turbocharger with VATN can produce very significant secondary improvements in fuel economy.

3. Fuel economy improvements are almost proportional to drive ratio changes. Performance must be regained through a greater degree of turbocharging.

F. General Conclusions

An advanced technology turbocharger can be developed that will be considerably more effective than present technology turbochargers for improving fuel consumption and optimizing emissions for automotive diesel engines. The turbocharger would weigh about 65 percent of present wastegate turbochargers. It would not be dependent on engine oil for lubrication. The cost is approximately equal to present wastegate turbochargers.

VII. RECOMMENDATIONS

The following areas need to be investigated to fully demonstrate the concept and its potential as well as answer any questions concerning mechanical integrity.

A. Conduct analytical studies and a design effort to define a heat transfer barrier system that will minimize the transfer of heat from the turbine to the compressor.

B. Conduct analytical studies and design efforts to aerodynamically rematch (using present hardware) the turbocharger to the John Deere diesel engine. This would include provision for additional turbine nozzle travel.

C. Build a turbocharger with the defined components from A and B above, apply the turbocharger to the John Deere engine and re-run all fuel consumption and emissions tests.

D. Extend the mathematical model to include emissions prediction.

E. Define and carry out a program to define the balancing tolerance of the individual rotating components as well as the assembled rotor.

F. Define and carry out a program that will result in a definition of the tolerance range for the major variables within the bearing system.

G. Develop a program aimed at utilizing the VATN to minimize adverse transient response effects of a turbocharged engine. This would include:

* A mathematical model to simulate the engine/turbocharger system that would show the effects of the VATN, bearing losses, intake and exhaust volumes, intake and exhaust temperatures and rotor inertia.

* Verification of analytical modeling techniques via dynamometer testing.

* Definition of an optimum engine/turbocharger system, including turbine nozzle and fueling scheduling.

* Manufacture and testing of the defined engine/turbocharger system.

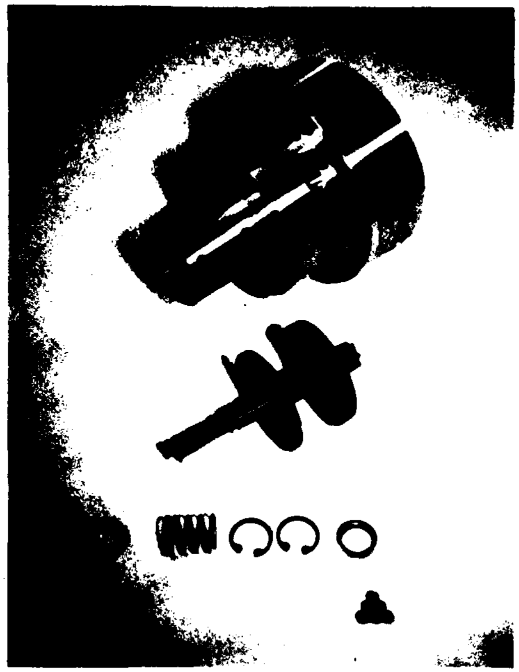
APPENDIX A

SIMULATED ROTOR TESTING

The simulated rotor test rig is shown on Figure A1. The rotor was machined from solid stock with the inner raceways being identical to the turbocharger as designed. Shaft diameters were the same as on the turbocharger. The two disks were located at the calculated centers of gravity of the compressor wheel and turbine wheel, and were of the same calculated masses and moments of inertia.

This rotor was housed in a bore identical to the turbocharger design with the same outer races and preload spring. It was driven by a small turbine on the end opposite the bearings.

The rotor was operated up to 125,000 RPM for a total of 1,000,000,000 revolutions (231.5 hours at average speed of 72651 RPM). A plot of the speed history is shown on Figure A2.



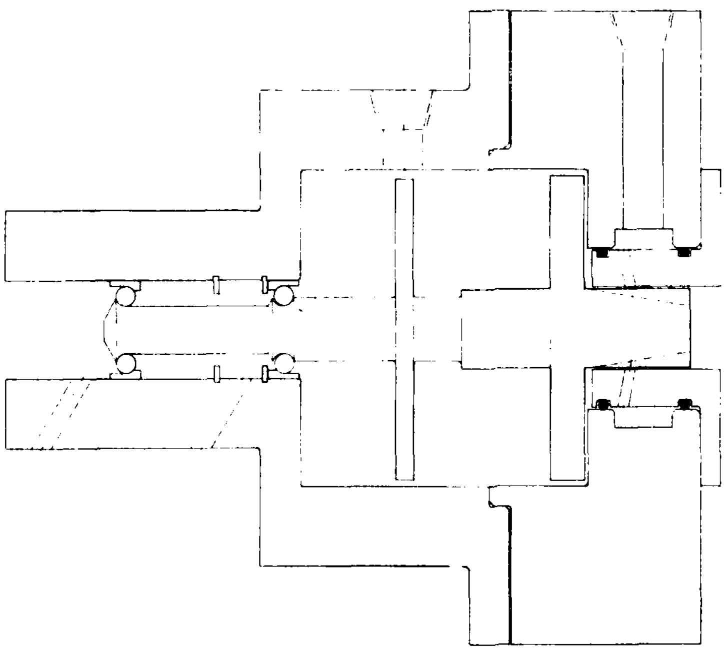


FIGURE A1 - SIMULATED ROTOR TEST RIG

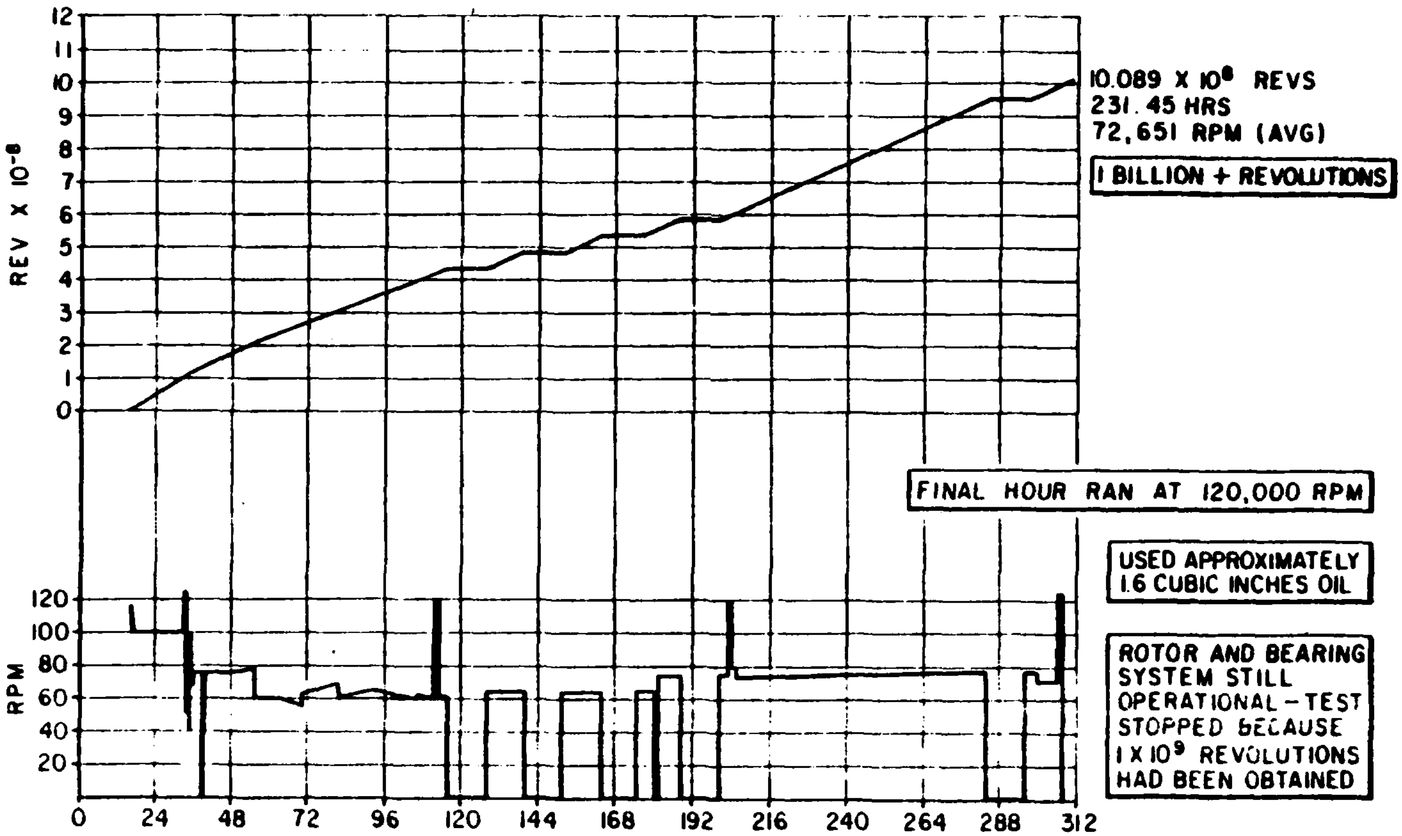


FIGURE A2 - SIMULATED ROTOR TEST - RPM Vs. TIME

APPENDIX B

OIL/WICK/WICK-SHAFT INTERFACE TESTS

Recirculation Allowed

Mobil DTE medium		Humble Turbo Oil #2380	
Wick #	Oil Consumed	Wick #	Oil Consumed
1	2.1 ml	7	1.7 ml
2	1.5 ml	8	2.8 ml
3	1.6 ml	9	.6 ml
4	2.1 ml	10	1.4 ml
5	.7 ml	11	.7 ml
6	1.2 ml	12	1.5 ml

No Recirculation Allowed

Mobil DTE medium		Humble Turbo Oil #2380	
Wick #	Oil Consumed	Wick #	Oil Consumed
3	6.8 ml	1	6.5 ml
4	8.6 ml	2	6.2 ml
6	7.4 ml	7	8.6 ml
11	9.6 ml	9	9.8 ml
12	8.8 ml	10	6.2 ml

APPENDIX C

TEST FACILITY

Hot gas, to drive the turbine, is provided by Cummins NH 250 diesel engine illustrated in Figure C1. A G-Power water brake dynamometer is utilized to load the engine and to control the exhaust temperature. The exhaust is discharged from the exhaust manifold into a 80 gallon tank to minimize pulsation as shown in Figure C2. From this tank the gases are directed through a six inch diameter pipe to a transition duct. This transition duct forms the passage from six inches diameter to the size and shape of the turbine inlet and provides for turbine inlet condition instrumentation and mounting of the turbocharger. The engine is in one room and the turbocharger is mounted in an adjacent room. A duct is provided at the turbine discharge to direct the gases from the turbocharger to a much larger vent duct, that is evacuated with a blower, leading to the roof. The turbine discharge duct incorporates manually operated butterfly valves (to control backpressure for Reynolds Number investigations) and provisions for turbine discharge condition instrumentation. The vent duct draws air from the test room (as well as turbine discharge gases) allowing fresh air from another vent to enter the room. A Meriam "laminar flow meter" is used to measure engine

FIGURE C1 - CUMMINS NH250 DIESEL ENGINE

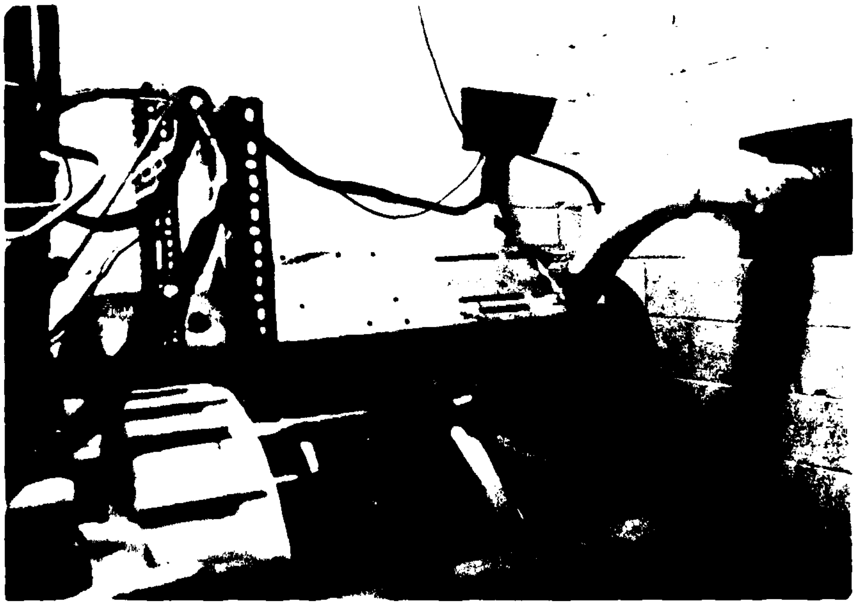


FIGURE C2 - EXHAUST PLENUM 80 GALLON TANK airflow (and therefore turbine airflow).

The compressor inlet ducting consists of a Meriam "laminar flow meter" followed by an eight inch diameter plenum (settling station) followed by a bellmouth entrance to the turbocharger and is shown in Figure C3. Compressor inlet condition instrumentation is incorporated in the plenum. The compressor exit ducting is illustrated in Figure C4 and consists of a short section of square tubing (with one end matching the compressor discharge size and shape) branching into three square tubes of different sizes. Each of these tubes contains a manually operated butterfly valve for controlling the pressure/airflow characteristics of the compressor.

A manually operated spur gear and lever system controls the position of the VATN.

The instrumentation used to measure overall performance is as follows:

* One Meriam model 50 MC2-4F laminar flowmeter including three thermocouples and two static pressure taps, to measure diesel engine airflow.
* Two static taps in the 80 gallon plenum to measure turbine inlet total pressure.

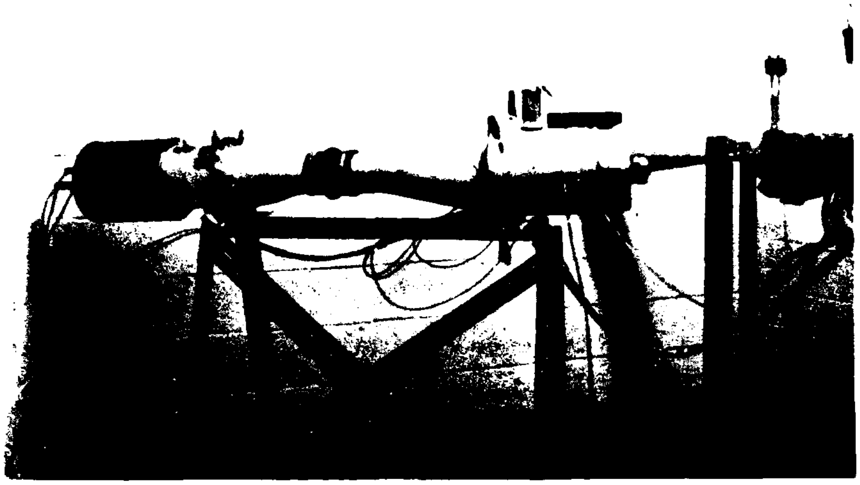


FIGURE C3 - COMPRESSOR INLET DUCTING

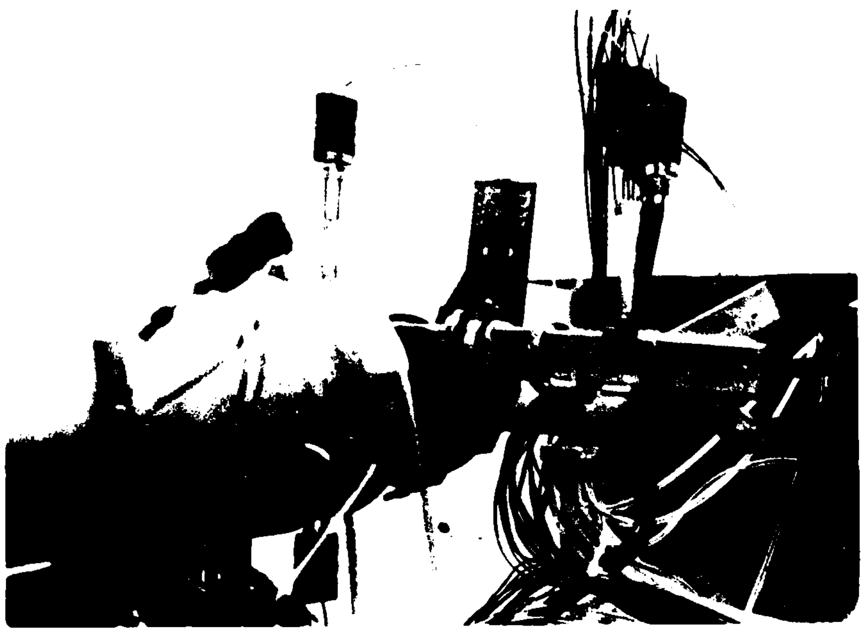


FIGURE C4 - COMPRESSOR EXIT DUCTING

* Four total thermocouples at the turbine inlet to measure turbine inlet total temperature.
* Nine total temperature thermocouples at the turbine exit to measure turbine discharge total temperature.
* One Meriam model 50 MC2-4F laminar flowmeter, including three thermocouples and two static pressure taps to measure compressor airflow.
* Four static pressure taps in the compressor inlet plenum to measure compressor inlet total pressure.
* Four thermocouples in the compressor inlet plenum to measure compressor inlet total temperature.
* Nine total pressure probes in the compressor discharge duct to measure compressor discharge total pressure.
* Four static pressure taps in the compressor discharge duct to measure compressor discharge static pressure.
* Nine total temperature thermocouples in the compressor discharge duct to measure compressor discharge total temperature.

A Hewlett-Packard 3052-A data acquisition system in conjunction with a 96 channel scanivalve is used to acquire the raw data and subsequently to perform calculations on the acquired data. The entire data acquisition system consists of:

HP 9825A desktop computer

HP 3495A scanner

HP 3455A highspeed digital voltmeter

HP 5301A counter

HP 9871A printer

Scanivalve-MSS2-48C9 multiple scanivale system

Calibration is conducted via the data acquisition system. All thermocouples were manufactured from the same lot of thermocouple wire for which calibration data was obtained (from 0^{o} F through 600^{o} F) from the supplier. This calibration data is programmed in the computer to correct the standard Hewlett-Packard subroutine for calculating temperature from thermocouple voltage. Additionally, a thermocouple calibration check program was written and is used prior to and following each test. This procedure entails obtaining a listing of all thermocouple calculated temperatures with the thermocouples immersed in both ice water and boiling water. Similarly, a compressor discharge total pressure probe check program was written and is used prior to and following each test. For this check the pressure rake is removed and placed in a special fixture. After a rapid increase in pressure (applied with a variator) each probe is "looked at" via the scanivalve and a listing of the pressure is obtained. The objective here is to ensure that all probes are responsive to pressure changes and no plugging exists. The pressure transducers (a low pressure 0-10 psi and a high pressure 0-100 psi) are calibrated via the data acquisition system, mercury manometers and a barometer. Manometer and barometer corrections for ambient temperature (both mercury and scale expansion) and latitude are included in the computer calibrations. Three reference pressures establish the slope of the pressure versus voltage line (ambient pressure and two manometer settings). Before these are established a zero voltage output is carefully set for ambient pressure. The accuracy of this system is well within:

* pressure readings accurate to less than .001 psi
* pressure sensitivity less than .0003 psi
* temperature reading accurate to less than $.25^{o}F$
* temperature sensitivity less than $.1^{o}F$

A program was written to monitor compressor performance (based on a small sampling of data), acquire data from all instrumentation when the desired stability had been achieved, and to calculate and print overall performance characteristics (including corrected and actual conditions). The data is collected in approximately 11 seconds and the results are printed in about 25 seconds from the initiation point.

A schematic of the test facility is shown on Figure C5.

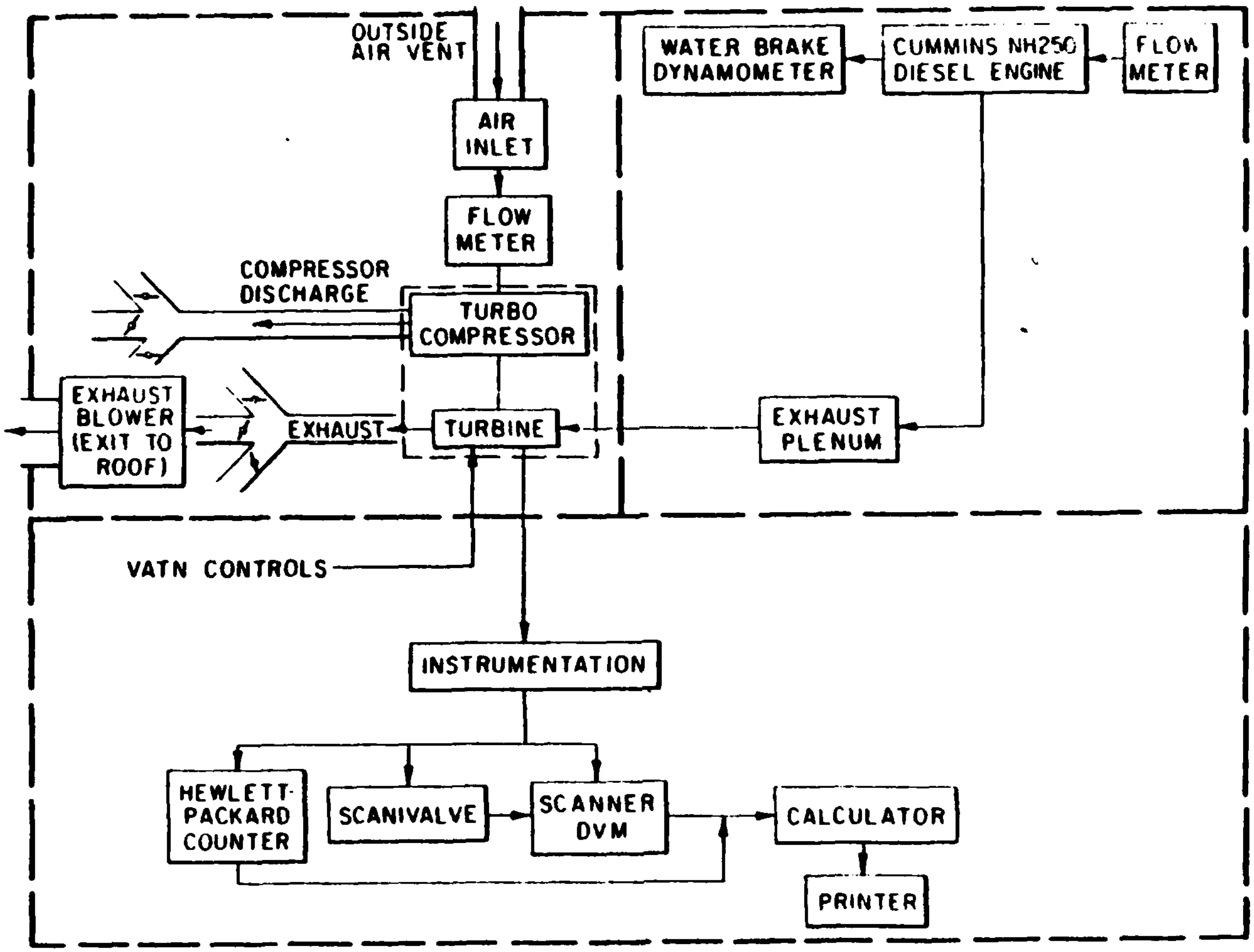


FIGURE C5 - TEST CELL FACILITY

APPENDIX "D"

FINAL REPORT

Submitted by

Southwest Research Institute

San Antonio, Texas 78284

TURBOCHARGING OF SMALL INTERNAL COMBUSTION ENGINE
AS A MEANS OF IMPROVING
ENGINE/APPLICATION SYSTEM FUEL ECONOMY

FINAL REPORT

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DECEMBER 1979

TABLE OF CONTENTS

		Page
	SUMMARY	1
I.	INTRODUCTION	3
II.	BASELINE TESTS	5
III.	TESTS WITH VARIABLE NOZZLE	12
	1. Maximum Power Output	12
	2. Part Load Tests	15
	3. Influence of Nozzle Position	15
	4. Comparison with Fixed Nozzle TC Results	17
	5. Transient Tests	20
IV.	EMISSIONS TESTS	23
	A. Conventional Emissions	23
	B. Smoke Tests	23
V.	DEVELOPMENT OF A MATHEMATICAL MODEL	28
	A. Model for Naturally Aspirated Engines	28
	B. Vehicular Application	31
VI.	MATHEMATICAL MODEL PREDICTIONS	33
	A. Development of Empirical Equations	33
	B. Model Prediction of Fuel Consumption in 13-Mode Federal Emission Cycles	34
	C. Model Predictions of the Fuel Ecomony in Vehicular Applications	34
VII.	CONCLUSIONS	42

APPENDIX A: CHOICE OF THE ENGINE
APPENDIX B: BASELINE TEST DATA AND RESULTS
APPENDIX C: MAXIMUM POWER OUTPUT TEST RESULTS - VARIABLE NOZZLE AREA TC
APPENDIX D: PART LOAD TEST RESULTS - VARIABLE NOZZLE AREA TC
APPENDIX E: EMISSIONS TEST DATA AND RESULTS
APPENDIX F: MATH MODEL DEVELOPMENT

LIST OF FIGURES

Figure No.		Page
1	VARIATION OF HIGH IDLE FUEL CONSUMPTION AND BSFC WITH RESPECT TO BREAK-IN TIME.	6
2	BRAKE SPECIFIC FUEL CONSUMPTION AND MAXIMUM POWER OUTPUT AT VARIOUS SPEEDS	8
3	BRAKE SPECIFIC FUEL CONSUMPTION AT VARIOUS LOADS FOR TURBOCHARGED AND NATURALLY-ASPIRATED ENGINES	9
4	VARIATION OF FULL LOAD VOLUMETRIC EFFICIENCY WITH RESPECT TO SPEED	10
5	COMPRESSOR ISENTROPIC EFFICIENCY AND PRESSURE BOOST AT VARIOUS LOADS AND SPEEDS	11
6	MAXIMUM POWER OUTPUT, BMEP, BSFC, PRESSURE BOOST AND EXHAUST - INTAKE PRESSURE RATIOS AT VARIOUS SPEEDS - FUEL RATE THE SAME FOR EACH NOZZLE LEVER POSITION AT A GIVEN SPEED.	13
7	AIR RATE, AIR-FUEL RATIO, AND TURBOCHARGER ROTOR SPEEDS AT MAXIMUM POWER OUTPUT CONDITIONS - FUEL RATE THE SAME FOR EACH NOZZLE POSITION AT A GIVEN SPEED.	14
8	BRAKE SPECIFIC FUEL CONSUMPTION WITH AiRESEARCH AND AERODYNE TURBOCHARGERS AT VARIOUS LOADS AND SPEEDS . .	18
9	AIR FLOW RATE WITH AiRESEARCH AND AERODYNE TURBOCHARGERS AT PART LOAD CONDITIONS.	19
10	SPEED RESPONSE OF THE ENGINE FOR A STEP INCREASE IN FUEL INPUT WITH THE AERODYNE TURBOCHARGER.	21
11	FUEL CONSUMPTION OF THE ENGINE FOR A STEP INCREASE IN LOAD WITH THE AERODYNE TURBOCHARGER.	22
12	EMISSIONS AND FUEL ECONOMY OVER 13-MODE FEDERAL DIESEL EMISSION CYCLE.	24
13	BAR PLOTS OF SMOKE TEST RESULTS.	26
14	INFLUENCE OF AIR-FUEL RATIO ON SMOKE (ALL CONDITIONS).	27
15	COMPARISON OF ESTIMATED AND EXPERIMENTALLY DETERMINED FUEL CONSUMPTION RATES FOR CAT 3208 ENGINE - HEAVY DUTY 13-MODE TEST CYCLE.	29

List of Figures, cont'd. Page

16 COMPARISON OF ESTIMATED AND EXPERIMENTALLY DETERMINED FUEL CONSUMPTION RATES FOR HINO EH700E ENGINE - HEAVY DUTY 13-MODE CYCLE. 30

17 COMPARISON OF MODEL PREDICTIONS WITH THOSE OF EXPERIMENTALLY OBTAINED FUEL FLOW RATES NATURALLY ASPIRATED JOHN DEERE ENGINE, MODEL 4239T. 35

18 COMPARISON OF MODEL PREDICTIONS WITH THOSE OF EXPERIMENTALLY OBTAINED FUEL FLOW RATES. JOHN DEERE, MODEL 4239T, WITH AERODYNE TURBOCHARGER, +10° NOZZLE LEVER POSITION. 36

19 COMPARISON OF MODEL PREDICTIONS WITH THOSE OF EXPERIMENTALLY OBTAINED FUEL FLOW RATES. JOHN DEERE, MODEL 4239T, WITH AERODYNE TURBOCHARGER, -10° NOZZLE LEVER POSITION. 37

20 MODEL ESTIMATION OF FUEL FOR VARIOUS N/V RATIOS 40

SUMMARY

The objective of this program was to evaluate the performance characteristics of a variable nozzle area turbocharger developed by Aerodyne Dallas. This turbocharger was tested on a small diesel engine (John Deere Model 4239T) in a test cell at Southwest Research Institute. This engine was originally equipped with a conventional turbocharger manufactured by AiResearch. The baseline tests were performed with this conventional turbocharger under several steady state conditions. The tests with the variable nozzle area turbocharger included steady state as well as transient operation. The emphasis was placed on power output, fuel economy and exhaust emissions.

The regulated exhaust emissions were determined over the Federal 13-mode Diesel Emission cycle. The measurements of smoke in percent opacity were made at eight different steady state conditions.

A mathematical model was developed in order to predict the fuel economy benefits of a variable nozzle area turbocharger in vehicular applications.

As expected, the turbocharger increased the maximum power output and significantly improved the fuel economy at full load conditions. However, at low speeds, the variable nozzle turbocharger did not consistently produce relatively high boost pressures; but the fuel economy with this turbocharger was significantly higher than that with the conventional turbocharger. This lack of consistency was probably due to difficulty in reproducing the same nozzle position under repeated conditions. The higher fuel economy (or lower BSFC) might be a result of the lower backpressure produced by the turbocharger.

The transient response of the engine did not significantly vary with changes in the nozzle area. However, the nozzles had enough control over the peak boost pressures so as to eliminate the need for a "waste gate" in the exhaust system.

As to the emissions, both turbochargers decreased hydrocarbons

and carbon monoxide and increased oxides of nitrogen. Also, smoke was reduced over the entire range of the engine. The variable nozzle turbocharger operating at one of its extreme ends (+ 10° nozzle position), was somewhat better in improving the fuel economy of the engine. This was attributed to the higher air-fuel ratios it maintained.

In general, the compressor efficiency of the variable area turbocharger was lower than that of the conventional turbocharger, indicating that this first generation turbocharger has room for further design improvements.

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Emissions
Variable Area Turbine Nozzles

20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

An advanced technology turbocharger featuring variable turbine nozzles, a broad operating range compressor, a ball bearing rotor system and a self contained lubrication system was manufactured, bench tested and applied to a 239 cid diesel engine. Fuel consumption and emissions data were collected during steady state engine performance testing. Compared to the "as received" turbocharged baseline engine the fuel consumption was improved at all conditions except medium speed/medium to high loads. Emissions were responsive to turbine nozzle position. Closed nozzles,

producing higher turbocharger speeds and intake manifold pressures, produced greater NO_2 and less CO, hydrocarbons and smoke than the baseline engine. Open nozzles produced the opposite results.

Analysis of the data reveal several areas for potential improvement. Both the compressor and turbine can be better matched to the engine as well as to each other. The transfer of heat from the turbine to the compressor can be reduced and additional nozzle travel can be utilized for further gains.

The general conclusion reached is that a fully developed advanced technology turbocharger can produce lower sfc, can flatten constant horsepower sfc versus engine speed characteristics and can be an effective control variable for emissions.

I. INTRODUCTION

The advantages of a turbocharger (TC) to an internal combustion engine are well known. It increases the maximum mean effective pressure and resultant maximum power output. It improves the specific fuel consumption by raising the mechanical efficiency under full load conditions. The fuel economy under part-load conditions can also be improved with a turbocharger by increasing the mean effective pressure and decreasing the engine speed without decreasing the desired power output. In spite of these benefits, the present-day turbocharger has limited pressure boost capabilities under low speed conditions. To overcome this difficulty, a turbine nozzle can be designed to improve the low speed pressure boost characteristics. If this route is taken a "waste gate", which wastes the exhaust and its energy, has to be provided at higher speeds and loads. In other words, one nozzle design is not adequate for optimum performance throughout the engine range. Therefore, a variable area nozzle was conceived to eliminate these problems. This type of TC can produce higher boost pressures at low speeds by changing the nozzle area and does not require a waste gate at high speeds. Aerodyne Dallas, as the prime contractor, developed such a turbocharger for use on small internal combustion engines. Southwest Research Institute, as a subcontractor, tested this turbocharger on an engine and determined its influence on fuel economy and emissions in a diesel engine under various loads and speeds. Also, the transient response with two different nozzle areas (positions) were examined. The results of this study are presented in this report.

The scope of this program was limited to testing the engine and turbocharger system in a test cell. In order to make predictions with respect to fuel economy in vehicular applications, some mathematical models were developed in this study and their results are also discussed here.

In the beginning of this program a diesel engine had to be selected to match the variable area nozzle turbocharger which was being developed by Aerodyne Dallas. An engine equipped with a conventional turbocharger was chosen with the intention of comparing the performance of this conventional TC with that of the test TC. The procedure followed for choosing this engine is described in Appendix A.

II. BASELINE TESTS

Before any testing was commenced, the engine was subjected to a systematic break-in. This consisted of operating the engine initially on a stepwise schedule from idle to full load and speed, and later running it at only steady conditions of rated load and speed. During the break-in, the operating variables were recorded at four hour intervals. In order to determine whether the break-in process was complete or not, the BSFC and high idle fuel consumption were plotted with respect to time, and this plot is shown in Figure 1. Although there is some scatter of results on these plots, it appeared that both BSFC and high idle fuel consumption reached a plateau at about 100 hours time after which the break-in was discontinued.

The baseline tests were performed under steady-state conditions with and without the production turbocharger at 1000, 1500, 2000, and 2500 rpm. The maximum load at each of these speeds was determined either by the upper limit of the fuel pump rack travel or by a chosen minimum air-fuel ratio of 20. The part loads at any one speed were set at 75, 50, and 25% of the full (maximum) load. The engine was operated at each of these speed-load combinations until the conditions were stabilized and various measurements were made to determine the performance characteristics of both the engine and the turbocharger. The recorded data and computed results of these tests are shown in Appendix B. The properties of the fuel supplied to the engine are also included in this appendix.

Also the conventional emission tests were performed with and without the production turbocharger and these will be discussed in a following section on emissions tests.

Performance Characteristics

The important results of these baseline tests were extracted from tables in Appendix B and are shown in Figures 2 through 5. The variation of power output and fuel economy is depicted in Figures 2 and

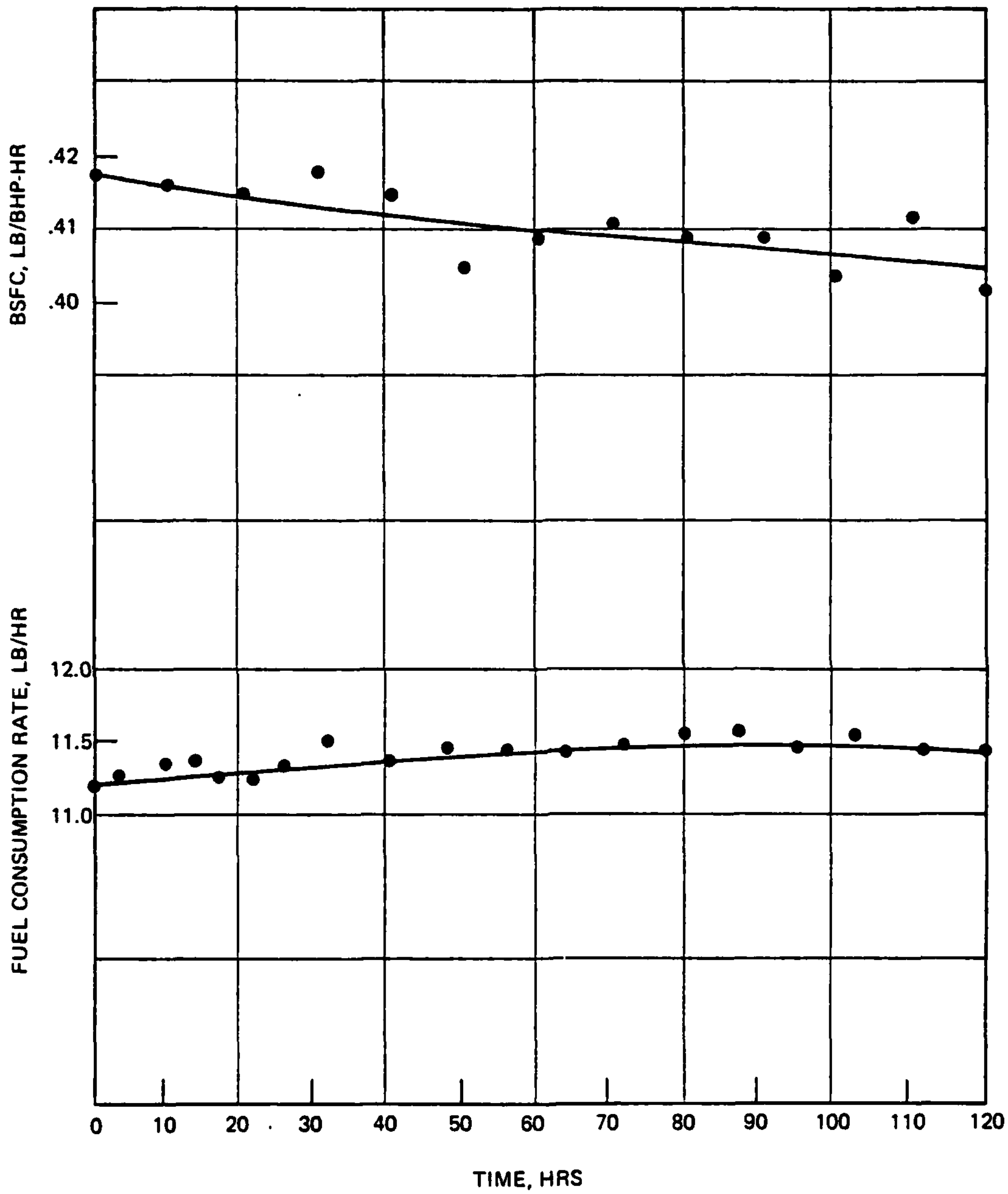

FIGURE 1 - VARIATION OF HIGH IDLE FUEL CONSUMPTION AND BSFC WITH RESPECT TO BREAK-IN TIME

3. These two figures indicate that at higher speeds, the turbocharger increased the power output of the engine quite significantly (almost 100%) and lowered the brake specific fuel consumption. The decrease in brake specific fuel consumption is mainly due to improvement in mechanical efficiency which is defined by

$$\eta_m = \frac{Bhp}{Bhp + Fhp}$$

where Bhp = brake horsepower

Fhp = friction horsepower

Under turbocharged conditions, the friction horsepower also increases, but not at fast as brake horsepower. If the friction horsepower increases as a lower rate than brake horsepower, the mechanical efficiency increases and thereby reduces the fuel consumption.

Another factor which also contributed to the lower specific fuel consumption was the decrease in fuel-air ratio with the turbocharger in operation. This decrease in fuel-air ratio improves the indicated thermal efficiency. At the engine conditions of interest, the increase in fuel economy is on the order of 2-5%. However, at lower speeds and lower loads, the turbocharger increased the brake specific fuel consumption, although the power output is slightly higher. This is probably due to relatively high exhaust backpressures at low load conditions. The slow rise in the maximum power output curve around 1500 rpm with the turbocharger is due to the limit set on air-fuel ratio (20).

As expected, the volumetric efficiency of the engine (Figure 4) decreased with increasing speed. However, the turbocharger reversed this trend above 1500 rpm.

Figure 5 indicates that the compressor isentropic efficiency and the pressure boost increased with speed and load, and the compressor efficiency was very low at low speed (1000 rpm) and low load (30 psi BMEP). These results and Figure 2 clearly indicate the limitations of the fixed geometry turbocharger.

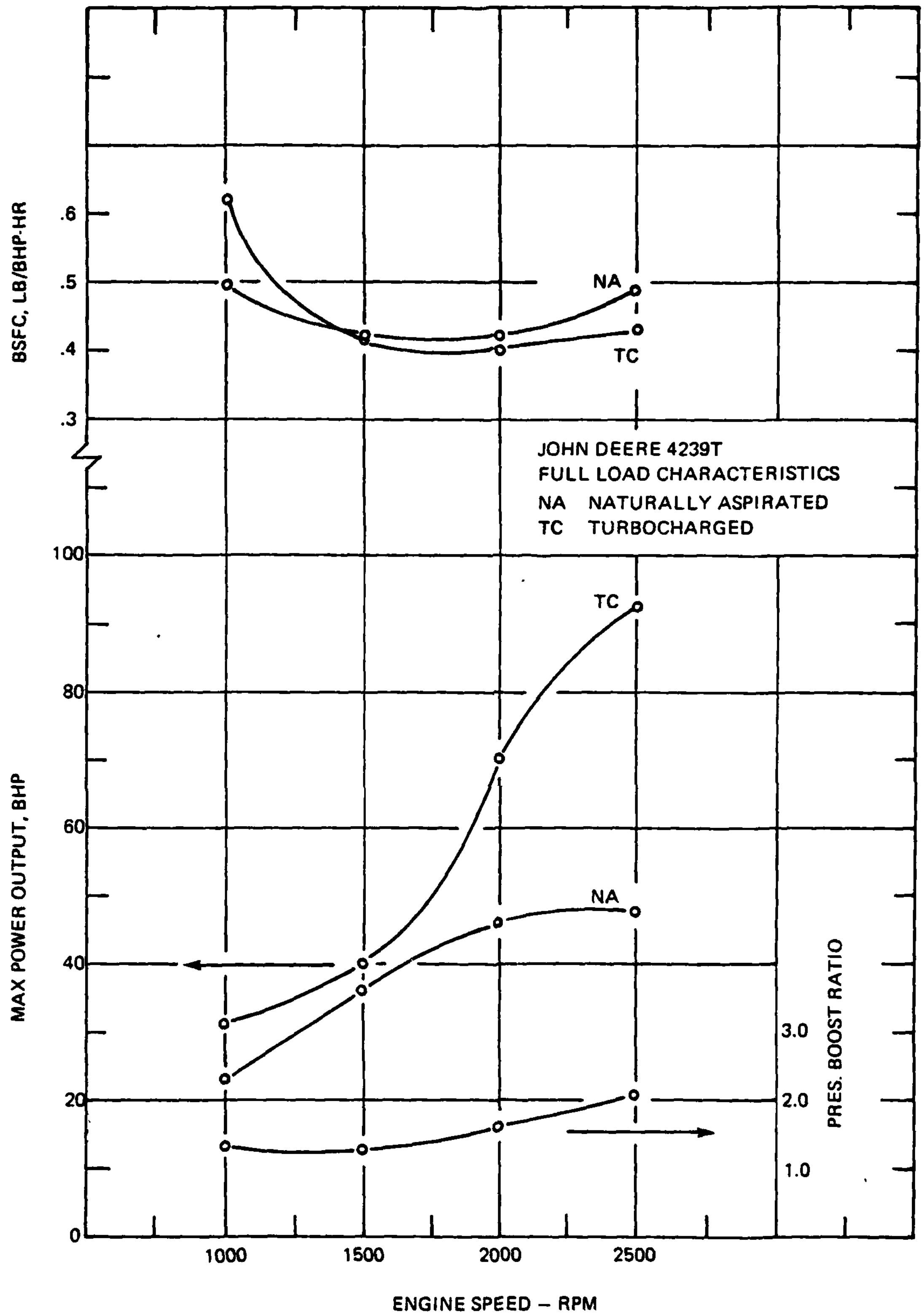

FIGURE 2 - BRAKE SPECIFIC FUEL CONSUMPTION AND MAXIMUM POWER OUTPUT AT VARIOUS SPEEDS

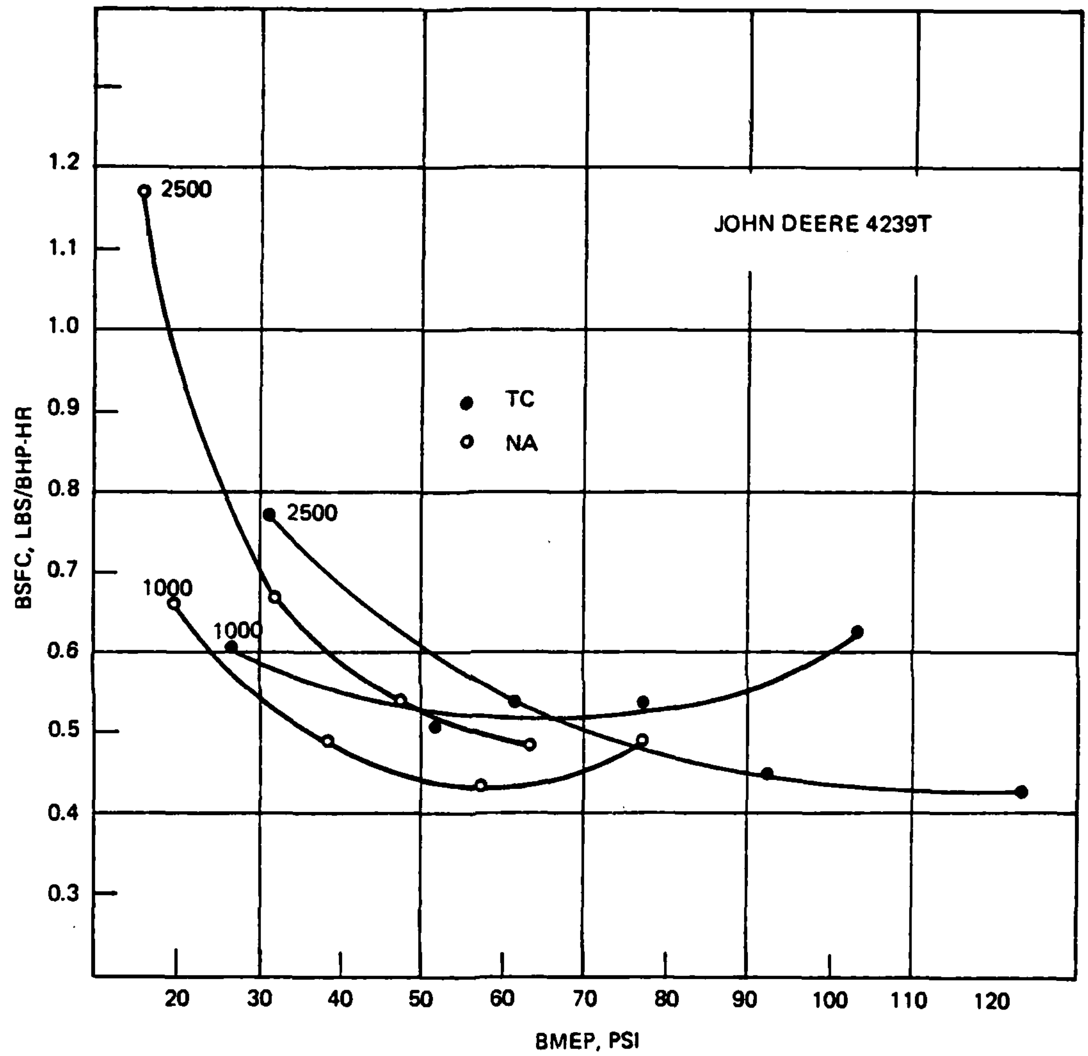

FIGURE 3 - BRAKE SPECIFIC FUEL CONSUMPTION AT VARIOUS LOADS FOR TURBOCHARGED AND NATURALLY-ASPIRATED ENGINES

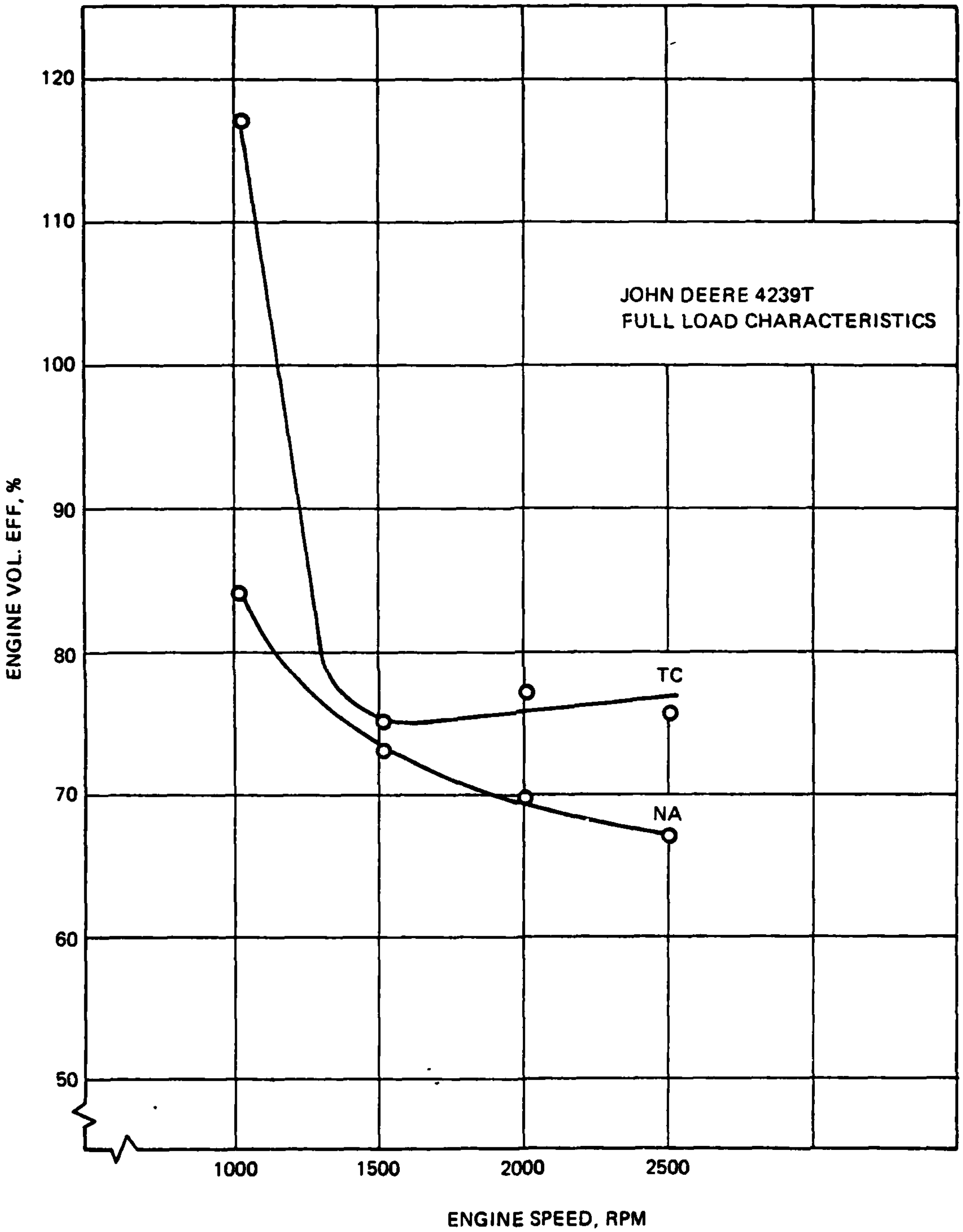

FIGURE 4 - VARIATION OF FULL LOAD VOLUMETRIC EFFICIENCY
WITH RESPECT TO SPEED

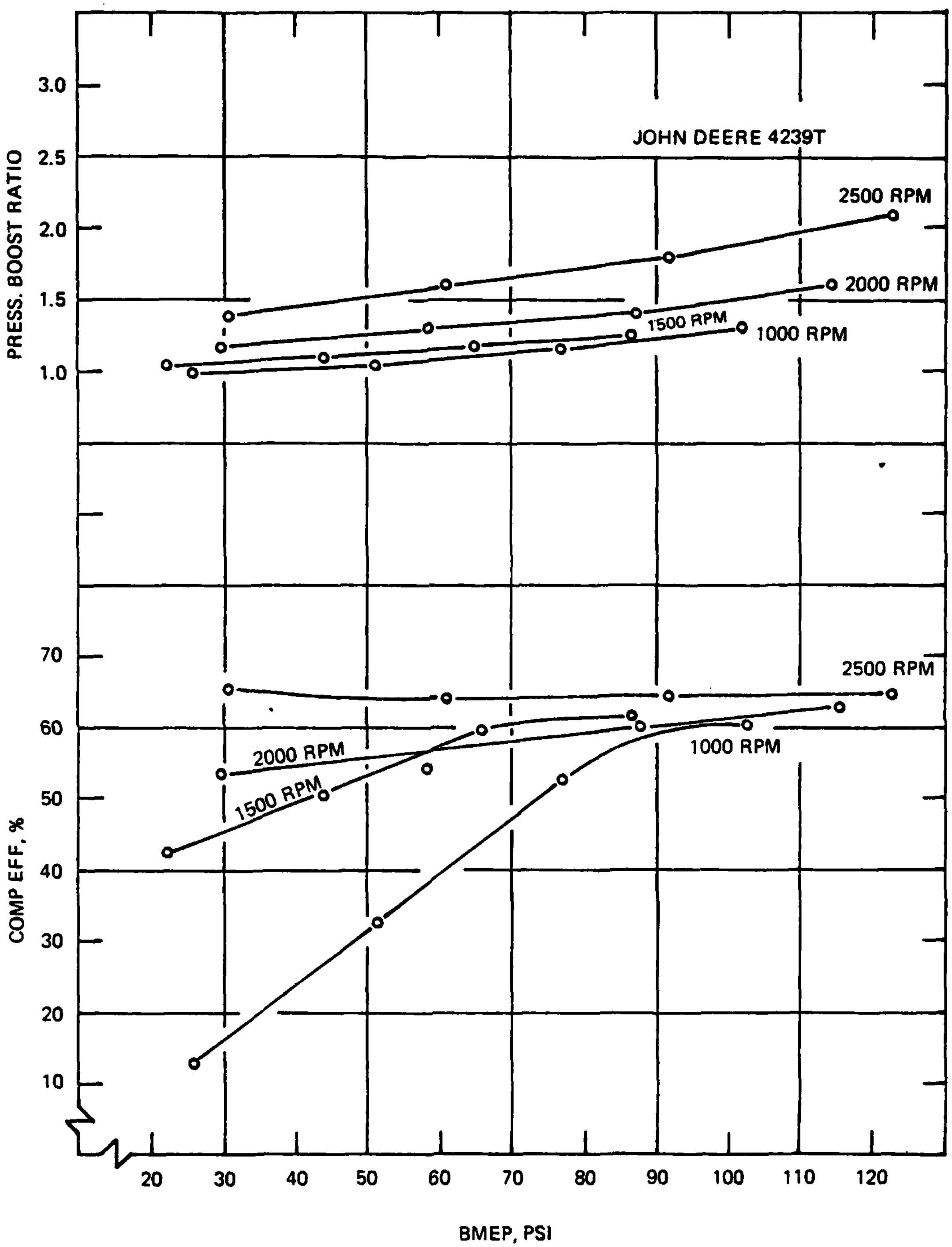

FIGURE 5 - COMPRESSOR ISENTROPIC EFFICIENCY AND PRESSURE BOOST AT VARIOUS LOADS AND SPEEDS

III. TESTS WITH VARIABLE NOZZLE

Turbocharger

The primary objective of these tests was to evaluate the performance characteristics of the Aerodyne turbocharger. This turbocharger was expected to yield higher boost pressures at low speeds, improve transient response, and produce more efficient control of peak boost pressures. In order to examine these features, a number of tests were conducted under steady state and transient conditions with different turbine nozzle positions (or areas). The steady state tests were further classified into maximum power output (full load) and part load tests. The maximum power output tests are discussed here first.

1. Maximum Power Output

The tests in this series were run between 750 and 2500 rpm with fuel rack position at maximum fuel delivery and turbine nozzle position at 0 and +10 degrees. The latter position set the nozzle area to a minimum. Altogether, a total of 12 tests were run, and the results are shown in Appendix C. The performance of the system was measured by the power output, brake specific fuel consumption, exhaust to intake pressure ratio, boost pressure, air flow rate, and air-fuel ratio. The foregoing variables, BMEP and turbine speed, are graphically shown in Figures 6 and 7.

The boost pressure, air flow rate, air-fuel ratio and turbine speed varied with nozzle position and were generally higher with +10 degrees setting at all speeds. However, in the cases of maximum power output, exhaust to intake pressure ratio and brake specific fuel consumption, the results were mixed. The power output and specific fuel consumption were better only at low speed with +10° setting. Also, the specific fuel consumption with both settings increased rapidly below 1000 rpm.

At higher speeds (above 1700 rpm) the decrease in

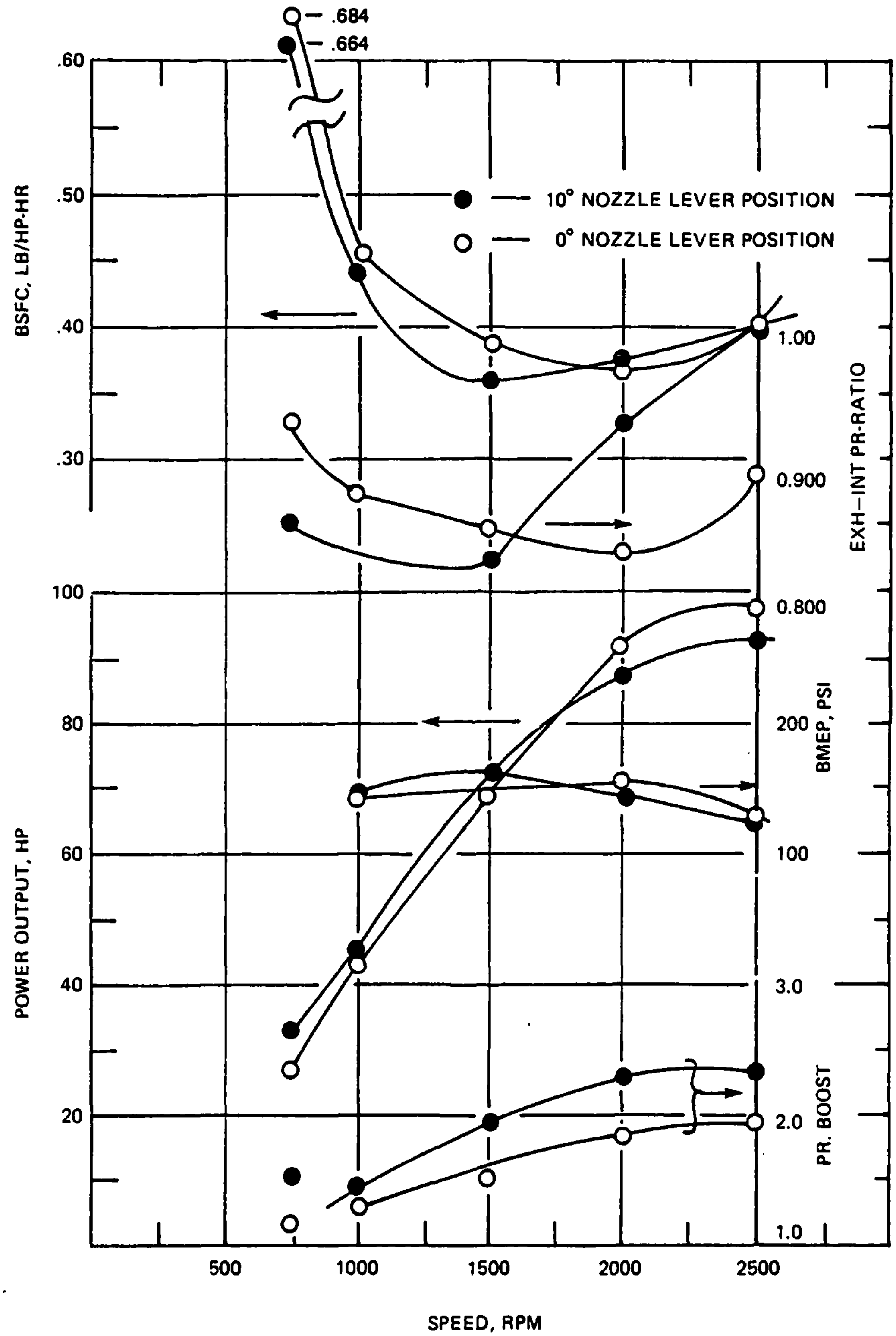

FIGURE 6 - MAXIMUM POWER OUTPUT, BMEP, BSFC, PRESSURE BOOST AND EXHAUST - INTAKE PRESSURE RATIOS AT VARIOUS SPEEDS - FUEL RATE THE SAME FOR EACH NOZZLE LEVER POSITION AT A GIVEN SPEED

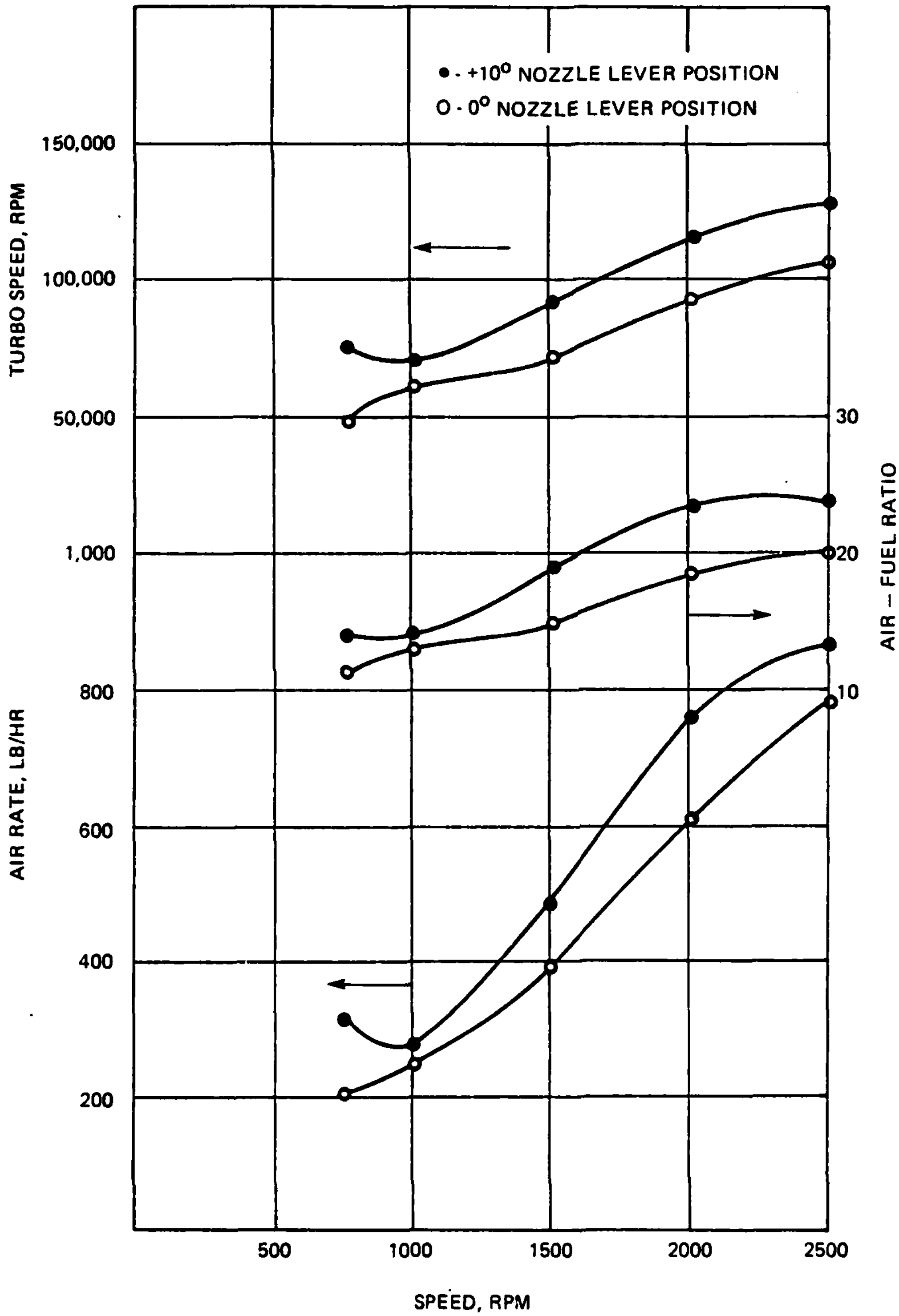

FIGURE 7 - AIR RATE, AIR-FUEL RATIO, AND TURBOCHARGER ROTOR SPEEDS AT MAXIMUM POWER OUTPUT CONDITIONS - FUEL RATE THE SAME FOR EACH NOZZLE POSITION AT A GIVEN SPEED

power output and fuel economy with +10° setting was probably due to two factors: 1) increase in exhaust to intake pressure ratio (Figure 6), which affects the frictional power loss, and 2) the constancy of the amount of fuel supplied to the engine in both nozzle positions.

2. Part Load Tests

Next, tests were performed under essentially the same load and speed conditions at which the baseline tests with the conventional turbocharger were conducted. These load-speed combinations are given below:

Speed RPM	Beam Load Lb.	Torque Lb-ft
1000	31.0	40.7
1000	61.0	88.0
1000	93.0	122.0
1000	124.0	162.8
1500	26.5	34.8
1500	53.0	69.6
1500	79.5	104.4
1500	106.0	139.2
2000	35.0	50.0
2000	70.5	92.6
2000	105.0	137.9
2000	140.0	183.8
2500	37.0	48.6
2500	73.5	96.5
2500	111.0	145.7
2500	148.0	194.3

3. Influence of Nozzle Position

The effect of nozzle position for each load-speed combination is graphically shown in a figure following each table. Shown in these figures (D-1 through D-16) are BSFC, air-fuel ratio, rate of fuel flow, air flow, compressor pressure boost, exhaust-intake pressure ratio, turbocharger rotor speed, and compressor isentropic efficiency. The results obtained in baseline tests with the AiResearch turbocharger are also indicated in these figures as bars originating from the ordinates. The isentropic efficiency was calculated using the following definition taken from Reference 1:

$$\eta_c \equiv \frac{T_1 \left\{ \left(\frac{P_2}{P_1} \right)^{0.285} - 1 \right\}}{T_2 - T_1}$$

where η_c = compression isentropic efficiency

T_1 = compressor inlet absolute temperature

T_2 = compressor outlet absolute temperature

P_1 = absolute pressure at compressor inlet

P_2 = absolute pressure at compressor outlet

In all tests, air flow rate, A/F, compressor efficiency, rotor speed, and compressor pressure boost varied with the nozzle position, and size of the variation depended on speed-load condition. The ratio of exhaust to intake pressure also varied slightly. A summary of brake specific fuel consumption results for all speeds and loads is shown in the following table:

Summary of Brake Specific Fuel Consumption Results

			Nozzle Position		
Speed RPM	Load Lb	BMEP psi	-8 and -10	0	+8 and +10
1000	31	25.7	.571	.574	.576
	61	50.6	.450	.448	.437
	93	77.1	.433	.430	.424
	124	102.8	.428	.422	.420

1500	26.5	22.0	.616	.632	.650
	53	44	.473	.471	.473
	79	65.5	.409	.412	.411
	106	87.9	.398	.397	.390
2000	35	29	.611	.615	.629
	70.5	58.5	.453	.449	.456
	105	87	.405	.404	.398
	140	116	.393	.375	.372
2500	37	30.7	.654	.653	.697
	73.5	61	.487	.491	.499
	111	92	.411	.404	.408
	148	122.7	.394	.395	.382

4. Comparison with Fixed Nozzle TC Results

The fuel economy (BSFC) and air flow rates determined for these two turbochargers under part-load conditions are summarized in Figures 8 and 9. These figures show that at low speeds the fuel economy with the Aerodyne turbocharger set at 0° nozzle position was significantly higher than that with the AiResearch turbocharger, and at higher speeds the differences were not significant. Similarly, one could compare the results of +10° nozzle position to those of AiResearch turbocharger.

Also, it was expected that the Aerodyne turbocharger would produce significantly higher boost pressure at lower speeds. But it did not produce this higher boost consistently with +10° nozzle position. This lack of consistency was probably due to difficulty in reproducing the same nozzle position. A related parameter, air flow rate, was also compared. At higher speeds (2000 rpm and above) the Aerodyne turbocharger, set at +10° nozzle position, had a higher flow rate than the conventional turbocharger, but at other conditions the flow rate was lower.

In view of the early development status of the Aerodyne turbocharger, it is best to judge its performance by comparing its overall performance against its own performance derived with a fixed nozzle position. The object of the foregoing comparison with the conventional (AiResearch) turbocharger was to explore whether there is any room for further improvement in the basic design of the Aerodyne turbocharger. It appears from the low speed test results discussed earlier that there is some room for further improvement since compressor

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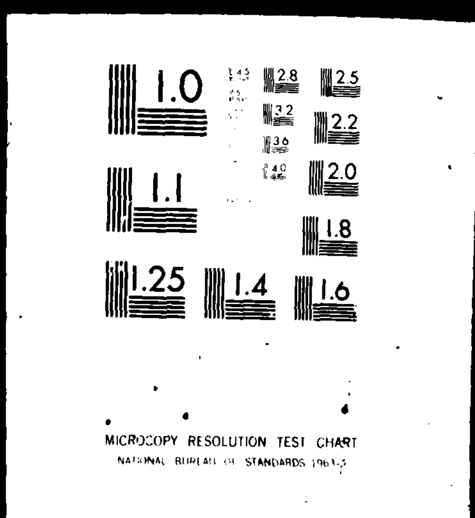


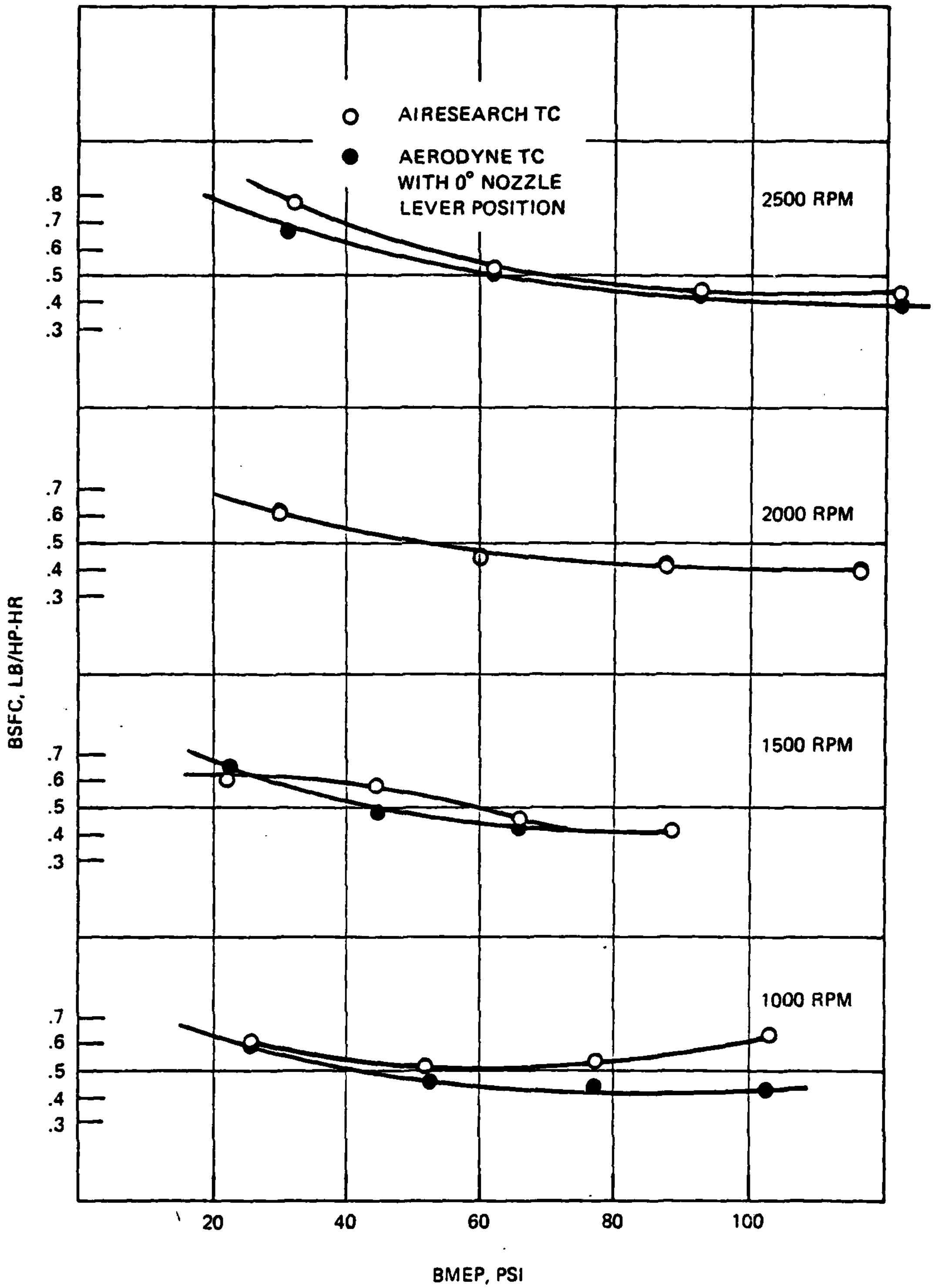


FIGURE 8 - BRAKE SPECIFIC FUEL CONSUMPTION WITH AiRSEARCH AND AERODYNE TURBOCHARGERS AT VARIOUS LOADS AND SPEEDS

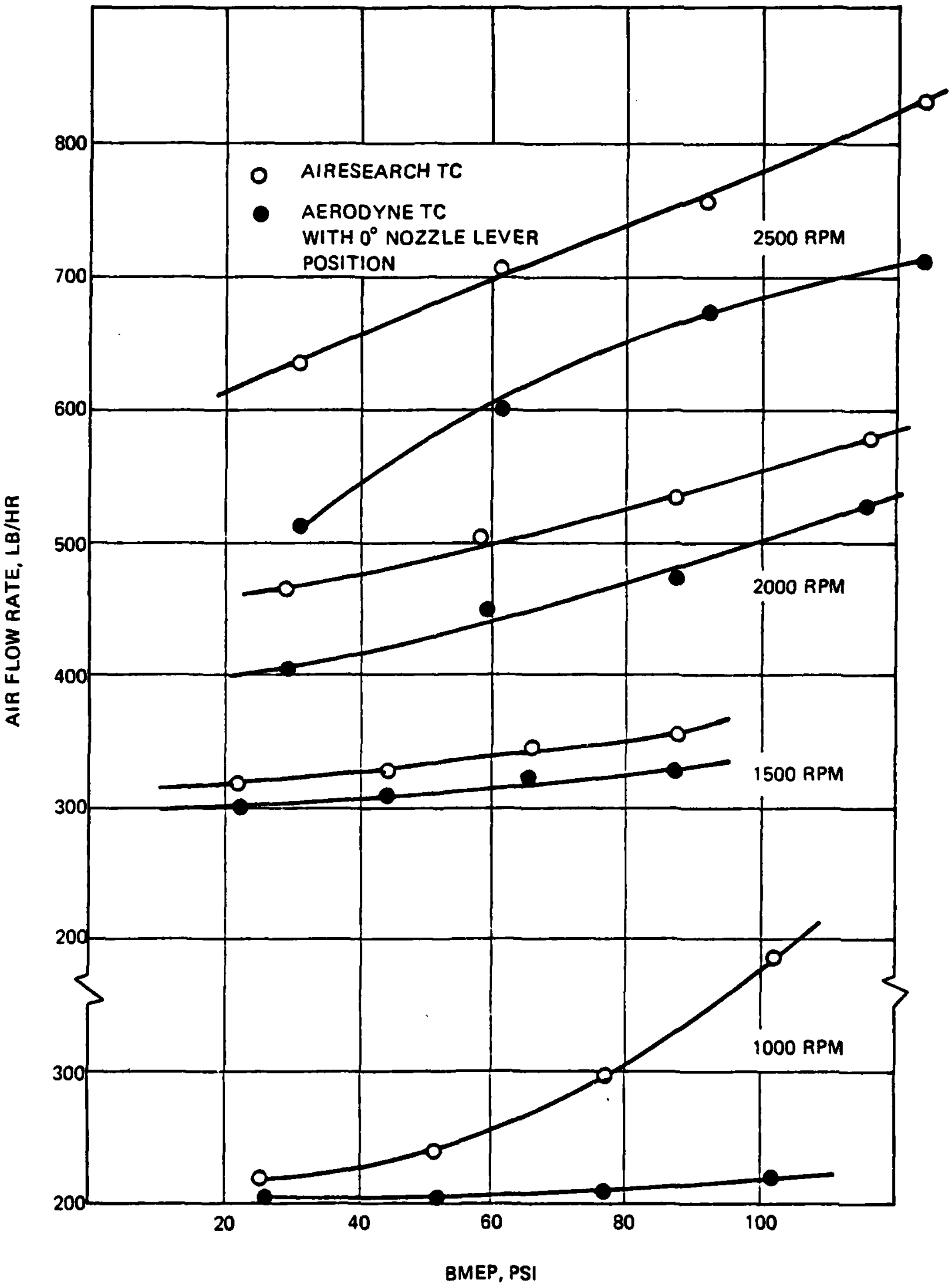


FIGURE 9 - AIR FLOW RATE WITH AiRESEARCH AND AERODYNE TURBOCHARGERS AT PART LOAD CONDITIONS

efficiency of the Aerodyne turbocharger was lower than the conventional. However, at 2500 rpm, and full load, the boost pressure ratio of the Aerodyne device could be controlled between 1.5 and 2.29 by varying the nozzle position. This range is probably adequate to eliminate the use of a "waste gate" under these conditions.

5. Transient Tests

In these tests, load, speed, and fuel flow were recorded using the instrumentation described in our previous report. Two different sets of tests were performed with nozzle settings at zero and 10°. In the first set, load was kept constant at 25 lbs. and fuel was supplied as a step function. The engine's speed response was recorded and is shown in Figure 10 for both settings. The response seems to be the same in both cases although one notices some delay for the fuel flow instrument to react.

In the second set of transient tests, the load was placed as step function and the response of fuel flow was recorded (Figure 11). Again, the differences in fuel flow response between zero and 10° nozzle settings were insignificant.

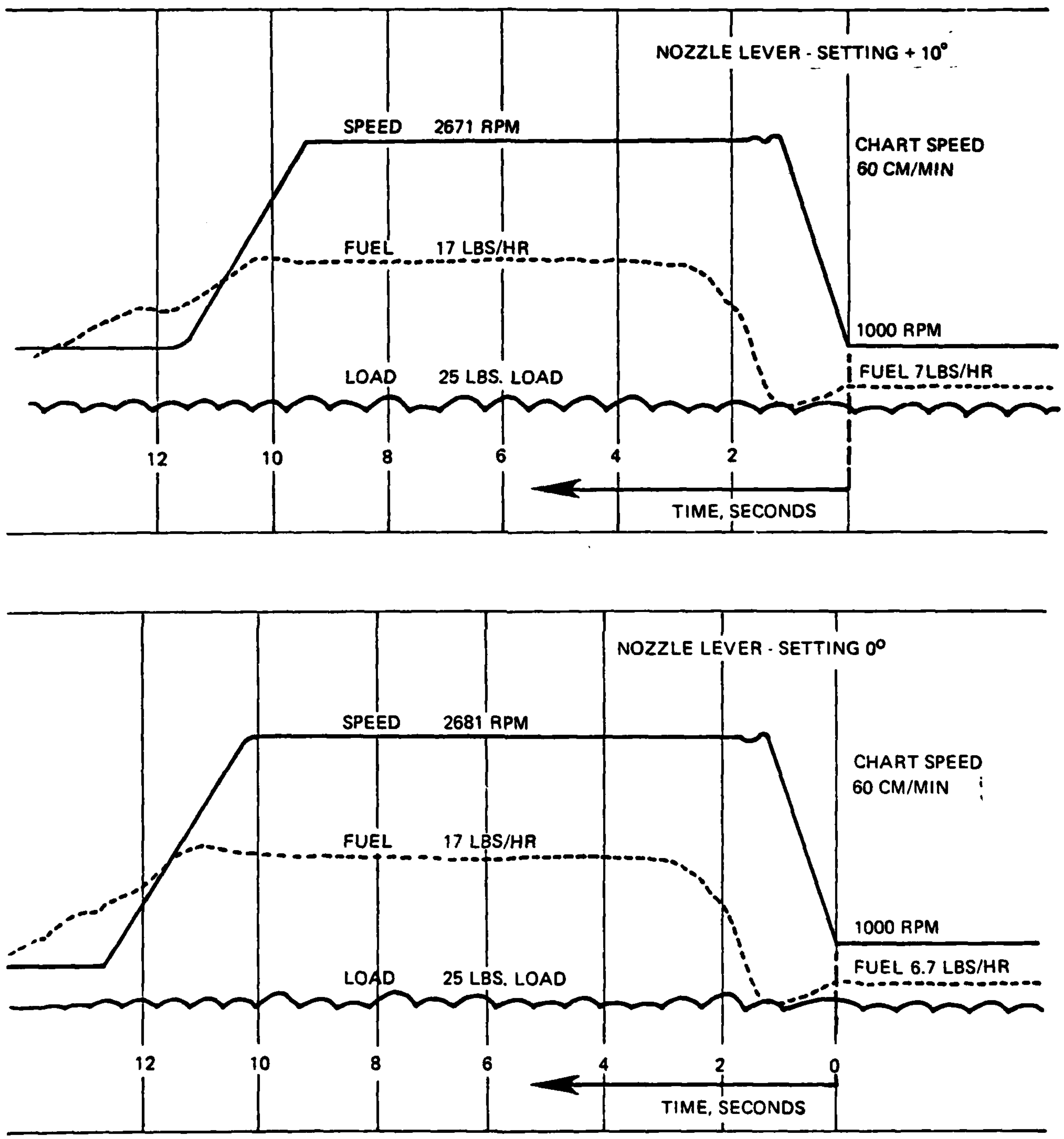


FIGURE 10 - SPEED RESPONSE OF THE ENGINE FOR A STEP INCREASE IN FUEL INPUT WITH THE AERODYNE TURBOCHARGER

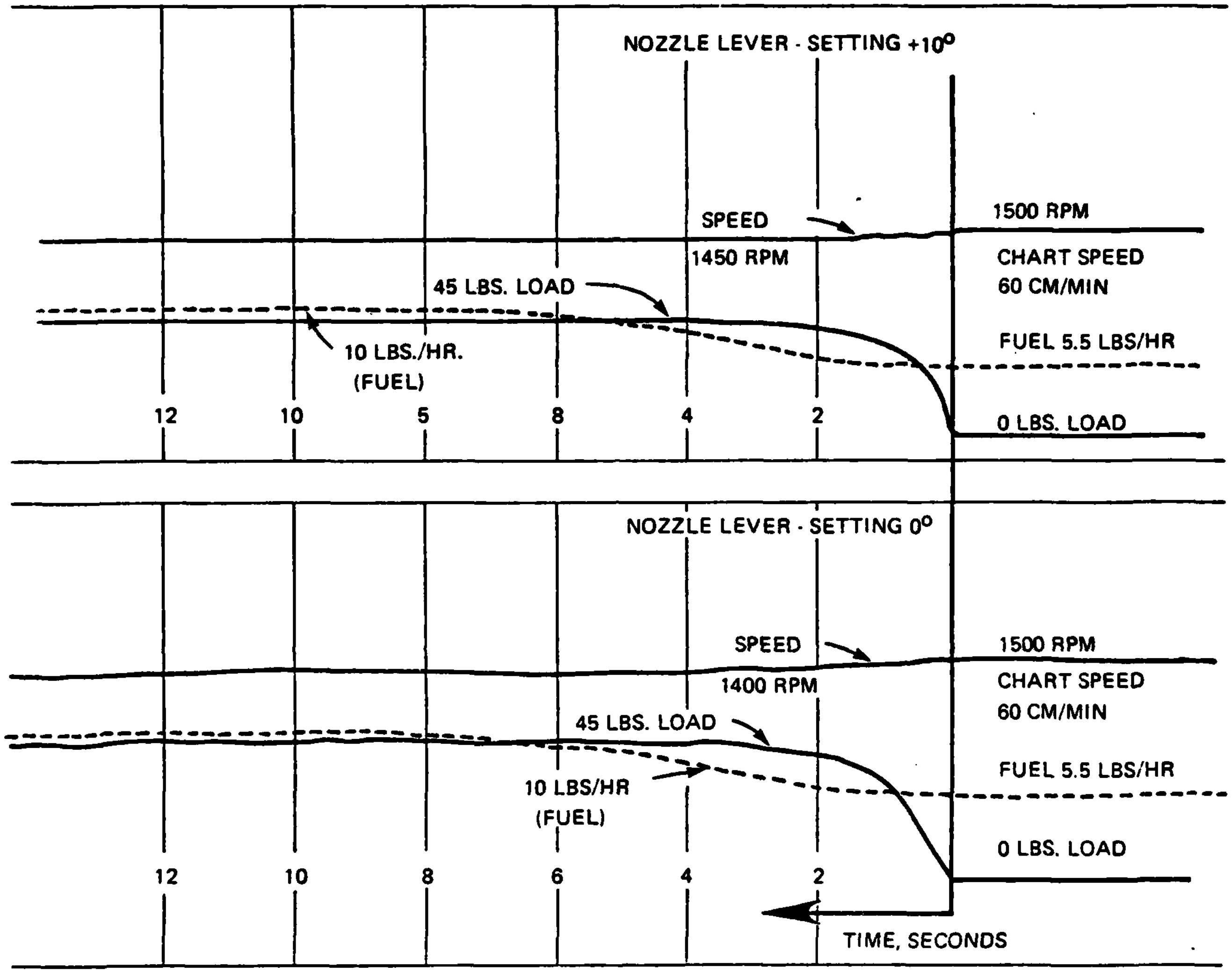


FIGURE 11 - FUEL CONSUMPTION OF THE ENGINE FOR A STEP INCREASE IN LOAD WITH THE AERODYNE TURBOCHARGER

IV. EMISSIONS TESTS

The Emission tests are divided into two classes: 1) conventional and 2) smoke tests. The conventional emissions include hydrocarbons (HC), carbon monoxide (CO), and oxides of nitrogen (NO_X). These results will be presented first.

A) Conventional Emissions

The baseline tests were conducted with and without the production turbocharger. In each case, the emissions were determined by the standard 13-mode Federal Diesel Emissions Test procedure. A speed-torque schedule of this procedure is shown in Appendix E. The speeds, loads, air-fuel ratios, and emissions recorded in these tests are also shown in this appendix. These results (averages) are shown in a bar plot in Figure 12. The influence of the turbochargers is quite vivid in this figure.

The oxides of nitrogen and the other two emissions (hydrocarbons and carbon monoxide) varied in opposite directions with and without the turbocharger. The hydrocarbon and carbon monoxide emissions were about 4 times higher without the turbocharger while the oxides of nitrogen decreased by twofold. The benefit of the turbocharger on fuel economy is about 16%. This trend appears to hold also with the variable nozzle turbocharger while the same differences are more pronounced with 10° nozzle position. However, difference in fuel economies between zero and 10° positions is negligibly small.

B) Smoke Tests

These tests were performed at full load and previous baseline test conditions. However, the lowest loads in the baseline tests were not considered for these tests because the air-fuel ratios under these conditions would be certainly higher than 50 and no smoke can be measured. Therefore, altogether a set of 16 different load-speed combinations were chosen and 44 tests were performed under steady

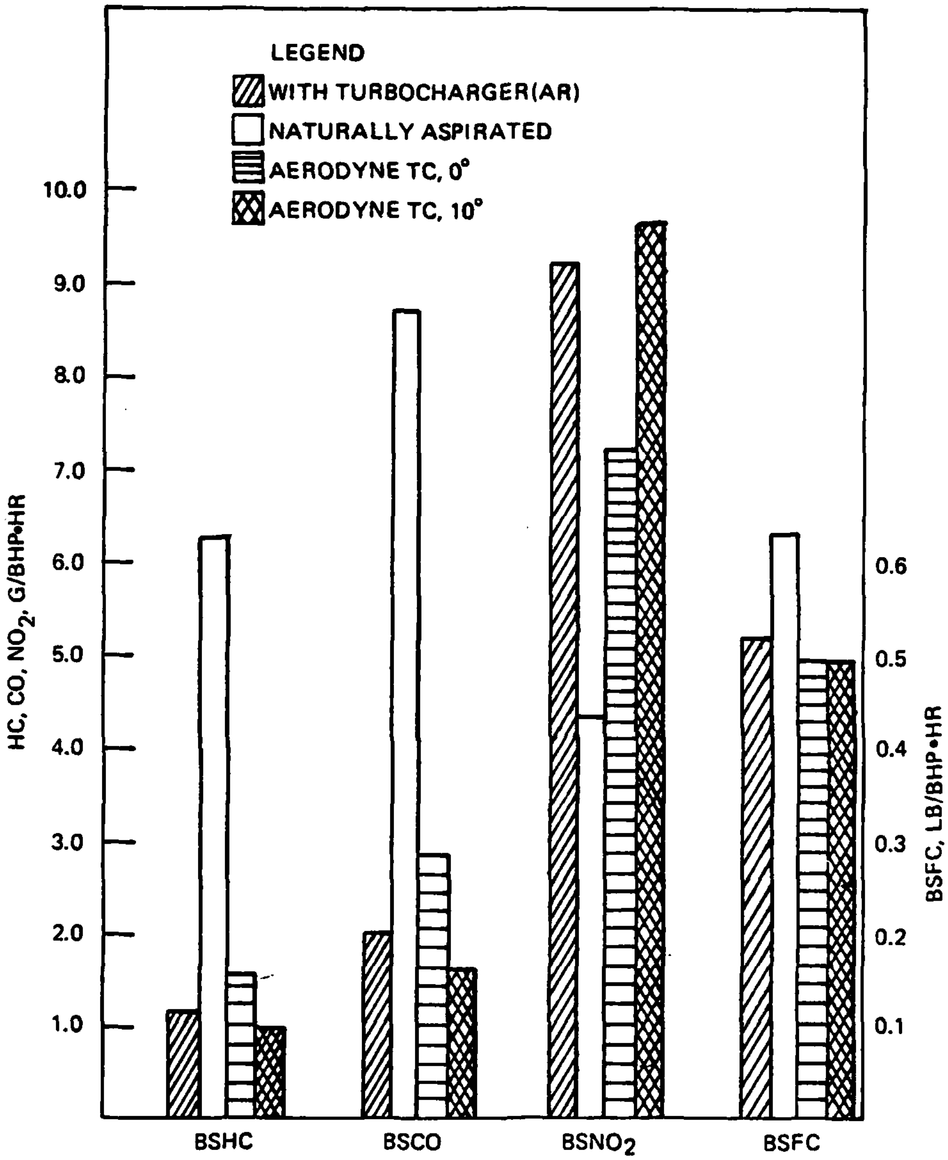


FIGURE 12 - EMISSIONS AND FUEL ECONOMY OVER 13-MODE FEDERAL DIESEL EMISSION CYCLE

conditions. These test conditions and the smoke results (in percent opacity) obtained are also shown in Appendix E. Along with the smoke, the air-fuel ratio is also listed. For comparison purposes among the turbochargers, the smoke results are shown in a bar plot in Figure 13. It appears that the smoke was lower with 10° position for which the air-fuel ratio was higher. To confirm this influence, the smoke in terms of percent opacity is plotted against the air-fuel ratio in Figure 14. The scatter in these results was probably due to repeatability which was about 1% at low levels and higher around 30% opacity. In general, the smoke decreased very rapidly up to an air-fuel ratio of about 25, and diminished slowly at higher air-fuel ratios. On the basis of these results, it can be concluded that the smoke is rigidly related to the air-fuel ratio regardless of load-speed condition and hardware. This is the reason why the fuel pump rack travel in baseline full load tests was controlled such that the air-fuel ratio would not fall below 20.

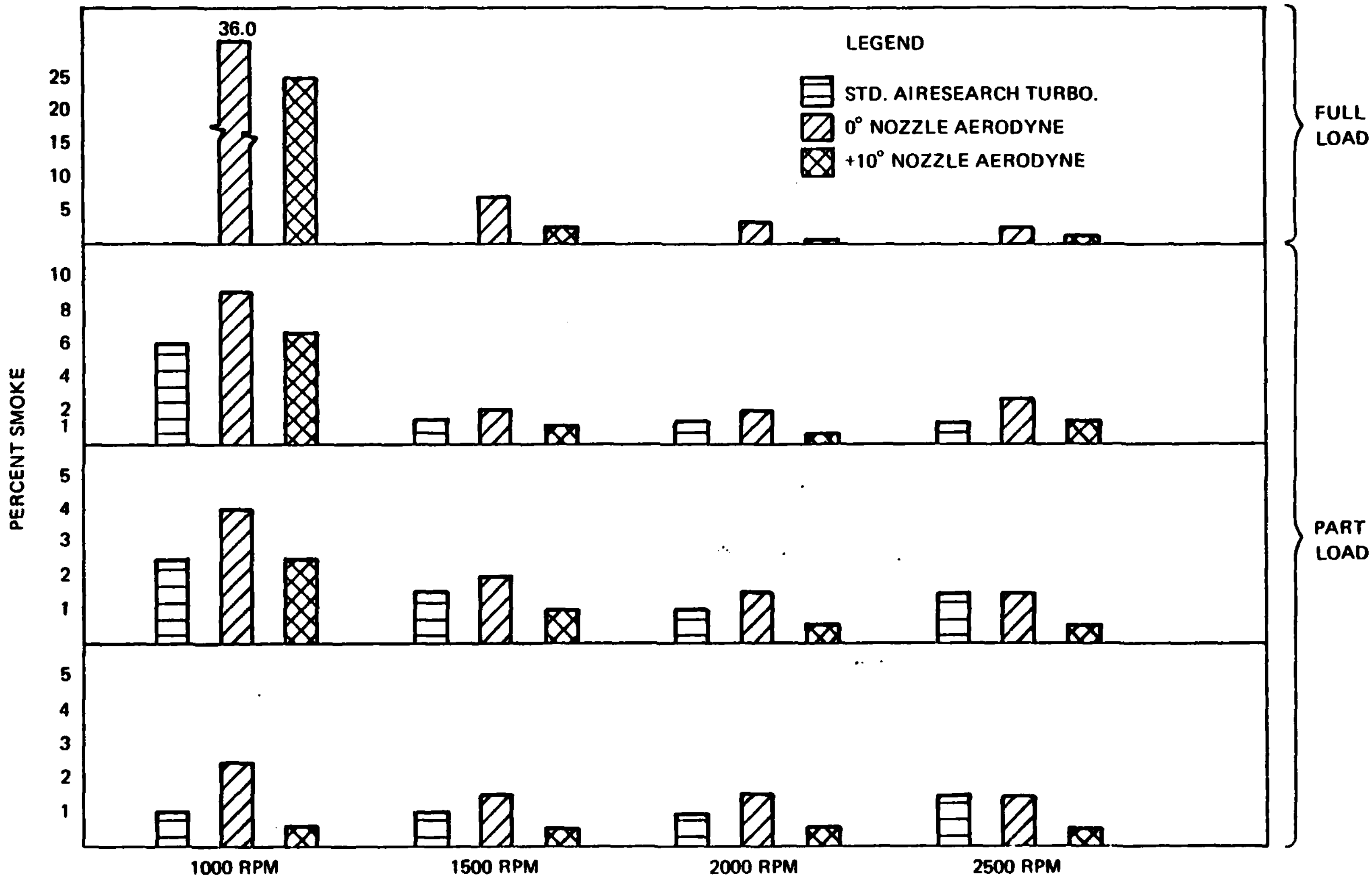


FIGURE 13 - BAR PLOTS OF SMOKE TEST RESULTS

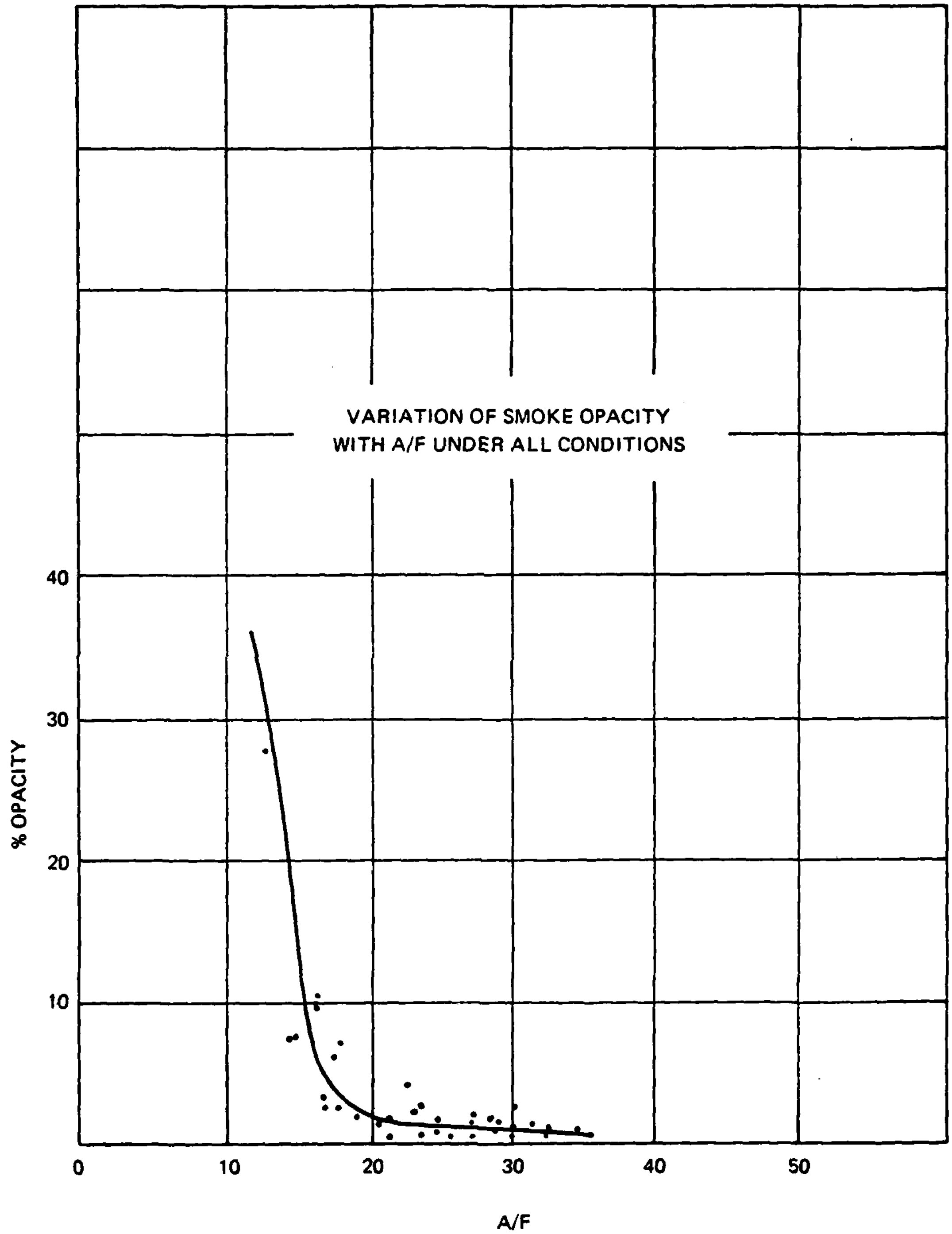


FIGURE 14 - INFLUENCE OF AIR-FUEL RATIO ON SMOKE
(ALL CONDITIONS)

V. DEVELOPMENT OF A MATHEMATICAL MODEL

In order to be able to predict the fuel economy and other operating characteristics of a diesel engine under steady state as well as transient conditions in vehicular applications, a mathematical model was developed for use with a computer. This model had provisions for incorporating the performance characteristics of a variable nozzle turbocharger. The model consisted primarily of a number of empirical formulae, which were derived from experimental data of various types of engines from Reference 1. Before this model was applied to the engines in vehicular applications, it was tested on naturally aspirated and turbocharged engines operated under steady state conditions representing the 13-Mode Federal Heavy Duty Engine Cycle. A computer flow chart of this model for the case of a naturally aspirated engine operated over the 13-Mode Federal Diesel Emission Cycle is shown in Appendix F.

A. Model for Naturally Aspirated Engines

Two different naturally aspirated, direct injection type engines, for which experimental data was available, were chosen to verify the model. These were the Caterpillar Model 3208 and the Hino Model EH700E. The engine specification data, atmospheric conditions, and lower heating value of the fuel were supplied to the computer as input. The calculated results are shown in Tables F-1 and F-2. The actual experimental results obtained at Southwest Research Institute for these engines are shown in Tables F-3 and F-4.

In addition to the fuel consumption, the mathematical model also computed brake horsepower (Bhp), air flow rate, fuel-air ratio, specific exhaust energy and cylinder pressure just before the exhaust valve opens. The tabulated fuel consumption results are also plotted in Figures 15 and 16 for comparison purposes. The model results differed from those of experiments by only an average of about 5%. The maximum difference is about 11% at high load (Mode 8) for the Hino engine.

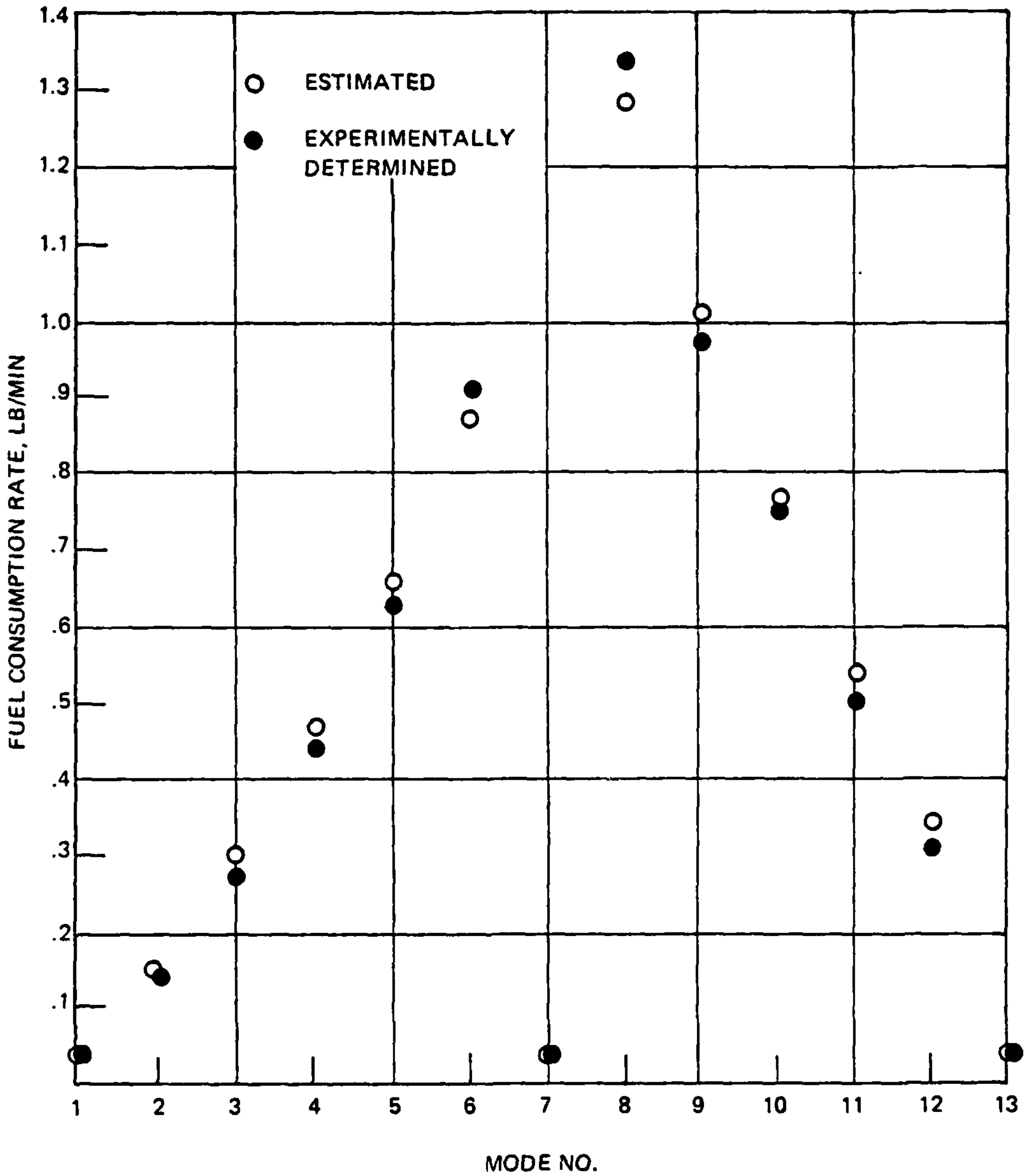


FIGURE 15 - COMPARISON OF ESTIMATED AND EXPERIMENTALLY DETERMINED FUEL CONSUMPTION RATES FOR CAT 3208 ENGINE - HEAVY DUTY 13 MODE TEST CYCLE

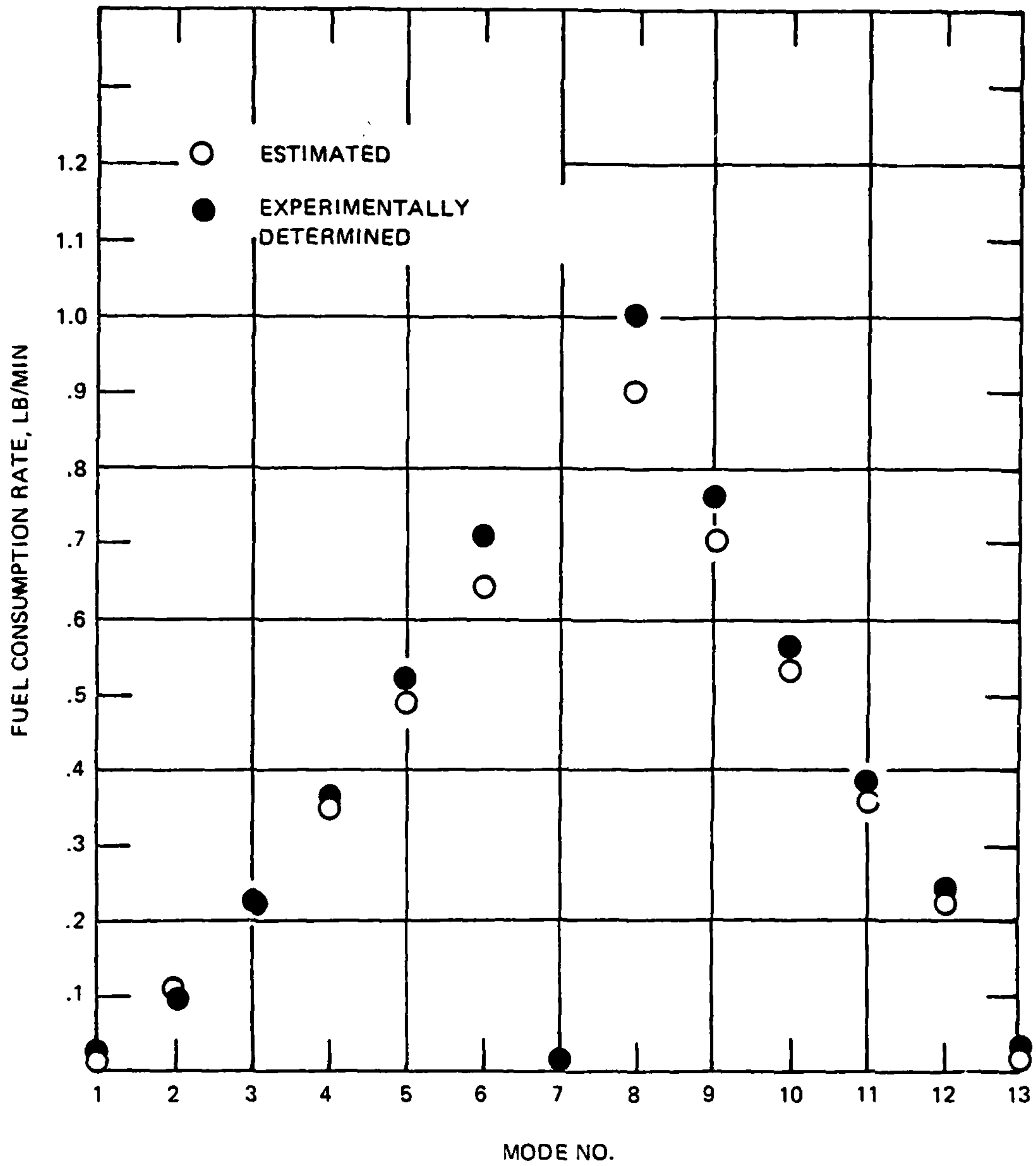


FIGURE 16 - COMPARISON OF ESTIMATED AND EXPERIMENTALLY DETERMINED FUEL CONSUMPTION RATES FOR HINO EH700E ENGINE - HEAVY DUTY 13 MODE CYCLE

An examination of Tables F-1 through F-4 shows that the computed and experimental fuel-air ratios are also very close to each other. Experimental results were not available to verify the specific exhaust energy and cylinder pressure (P4) shown in Tables F-1 and F-2.

B. Vehicular Application

With the agreement verified between the mathematical model and experimental results, the model was next applied to a small naturally aspirated diesel engine in vehicular use as a further check. The vehicle was a 1978 Volkswagen Rabbit. The measured fuel economy of this vehicle was available over the Light Duty Vehicle EPA cycle. The model was modified to include transmission gear ratios and drive shaft horsepower. The fuel consumed was estimated for each second of the cycle and the total consumption was determined by summing up over the entire cycle. A sample of the computer output is shown in Table F-5. The gear ratios and shift points used in this model are:

Gear	Ratio	Speed Range
1st Gear	3.45	0-15 mph
2nd Gear	1.94	15-25 mph
3rd Gear	1.37	25-40 mph
4th Gear (final)	1.10	40- mph
Rear End (or Axle) Ratio	3.90	
Wheel Revolutions/Mile	918	

Several equations for the drive shaft horsepower were tried, including the one experimentally determined on a chassis dynamometer for a 2250 lb. compact car at Southwest Research Institute. These equations are given below along with fuel economies obtained on the computer.

Equation	Computed Fuel Economy City mpg	Highway mpg
(1) $P = \frac{V}{375}\left(\frac{12W}{1000} + \frac{1.24AV^2}{1000}\right)$	69	63
(2) $P = 4.182 - .267V + \frac{1.1235V^2}{100} - 4.417V^3/10^5$	57	59
(3) $P = 4 + \frac{V}{338000}(8W + 1.1\ AV^2)$	48	55
(4) $P = 5 + \frac{V}{338000}(8W + 1.1AV^2)$	45	53

where V = speed, mph
W = weight, lb.
A = frontal area, ft^2
P = power, hp

Equation (1) = from Reference 2
Equation (2) = from unpublished Southwest Research Institute experimental results
Equation (3) = from Wood, C.D., and Hambright, R.N., "Assessment of Supercharging Systems for Gasoline Engines", Final Report prepared for IHI Industries, Tokyo, Japan, August 14, 1977
Equation (4) = Modified equation (3)

The corresponding fuel economies obtained by EPA in their actual tests were 40 and 53 mpg for City and Highway cycles, respectively. Our computed fuel economy for the City cycle is somewhat higher than that determined by EPA.

VI. MATHEMATICAL MODEL PREDICTIONS

The mathematical model which was employed earlier was further modified to accept the Aerodyne variable nozzle turbocharger. In this model, the intake manifold conditions were estimated by using the characteristics of the Aerodyne turbocharger, which were derived from the large number of tests we have conducted in this program. These characteristics were transformed into empirical equations and incorporated into the model.

A. Development of Empirical Equations

The important characteristics required for the model were those which could determine the conditions in the intake and exhaust manifolds. These were exhaust backpressure, compressor pressure boost, and temperature rise across the compressor. In order to evolve these characteristics in the form of empirical equations, the variables were plotted on several graphs in generalized form. These plots are shown in Appendix F (Figures F-2 through F-10). The variables were chosen such that these characteristics can be applied to any size engine.

Figures F-2 through F-4 depict the relationship between the pressure boost (Pi/Pn) and a function of BMEP and engine speed for all the nozzle positions. Similarly, the temperature rise across the compressor was plotted with respect to pressure boost in Figures F-5 and F-7. We attempted to develop the relations for temperature rise using the compressor efficiency and generalized engine variables. These efforts were not successful. Therefore, this temperature rise was plotted with respect to pressure boost. As can be seen from these figures, a fairly close relationship existed between these two variables.

The relationship between exhaust backpressure (Pe), intake manifold pressure (Pi) and BMEP is shown in Figures F-8 through F-10. Unlike in previous plots, a fairly linear relationship was found between Pe/Pi and BMEP. For all these plots empirical equations were developed with the help of a computer and shown on each plot. These equations were later incorporated in the math model to determine the intake and exhaust manifold conditions. The complete model was again tested against the results obtained in the 13-mode Federal Heavy Duty Engine Emission Cycle for the John Deere engine with the Aerodyne turbocharger.

B. Model Prediction of Fuel Consumption in 13-Mode Federal Emission Cycles

The experimental results for the 13-mode cycle obtained with the John Deere engine with the Aerodyne TC are shown in Tables E-4 through E-5. These were discussed in an earlier section. However, for comparison purposes, the results of fuel consumption were extracted from these tables and are shown along with the math model predictions in Figures 17 through 19. The model predictions for the naturally aspirated engine (Figure 17) differed by less than 8.1% from those of the experimental results except in Mode 12. In the case of turbocharged engine with +10° nozzle position (Figure 18) the maximum difference was about 8.7% in Mode 8. With zero degrees nozzle position (Figure 19) the agreement between the model predictions and experimental results was even better. Consequently, the empirical equations developed were considered to be satisfactory and were incorporated in the model to predict the fuel economy in vehicular applications. The results of this phase are discussed below.

C. Model Predictions of the Fuel Economy in Vehicular Applications

Earlier, the fuel economy of a 1978 VW Rabbit equipped with a naturally aspirated diesel engine was determined over both the Urban and Highway cycles. The same vehicle and engine combination was chosen again, and the fuel economy was estimated for various combinations of transmission gear ratios and nozzle positions. Because the engine with a turbocharger can produce higher power output, the gear ratios were reduced from those of the naturally aspirated case. The following table shows the effective ratios of engine speed (N) to vehicle speed (V) in different gears.

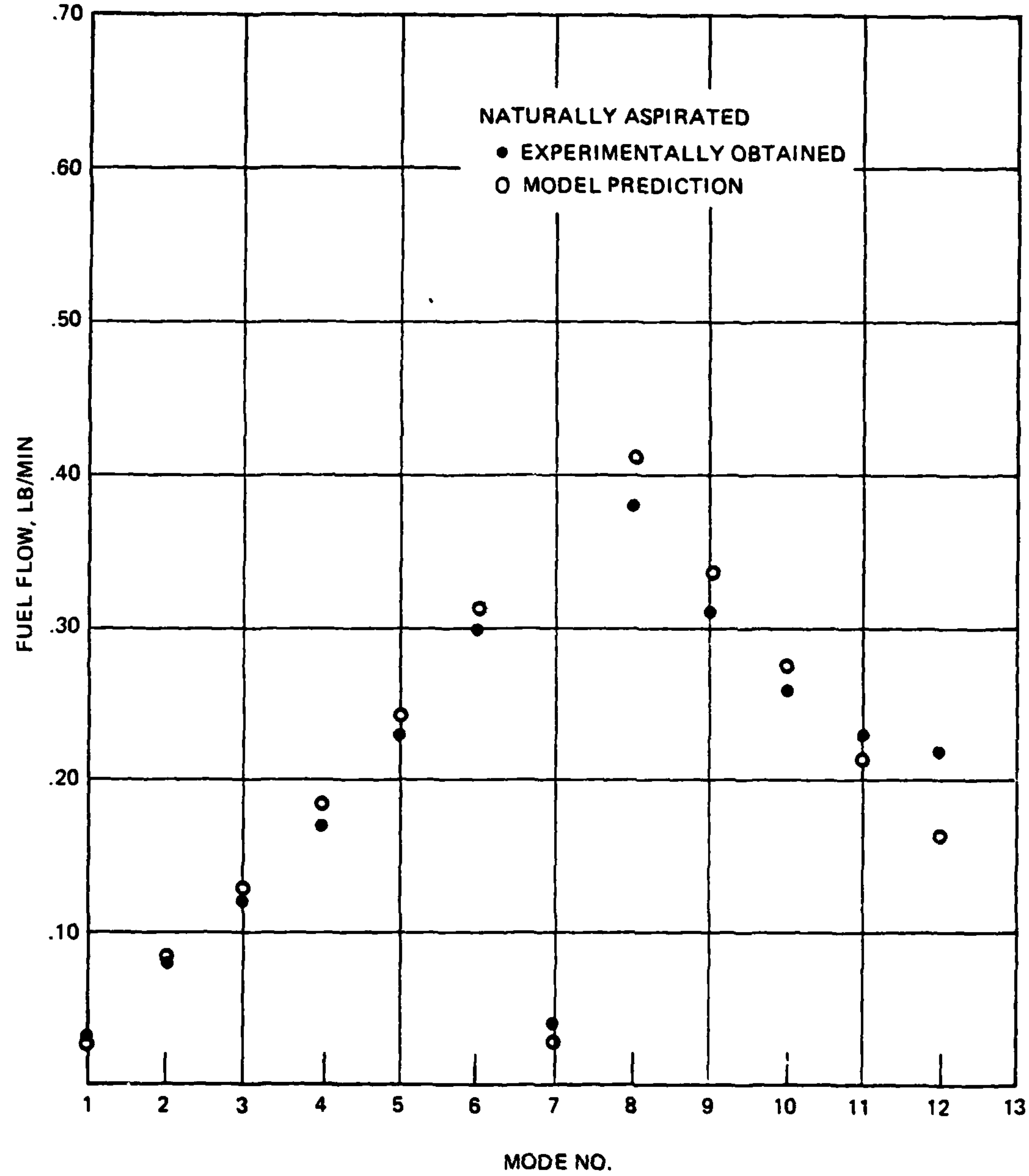


FIGURE 17 - COMPARISON OF MODEL PREDICTIONS WITH THOSE OF EXPERIMENTALLY OBTAINED FUEL FLOW RATES NATURALLY ASPIRATED JOHN DEERE ENGINE, MODEL 4239T

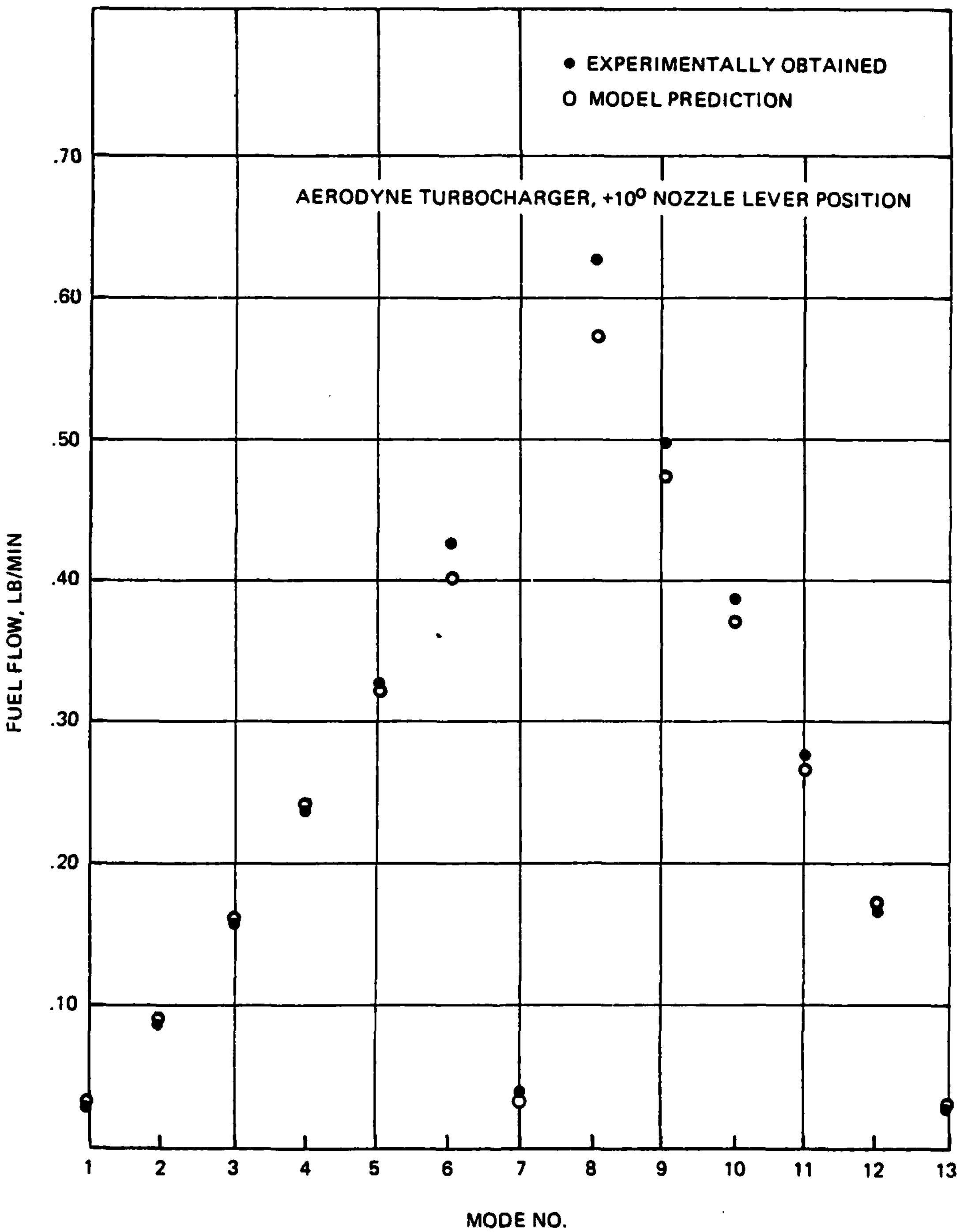


FIGURE 18 - COMPARISON OF MODEL PREDICTIONS WITH THOSE OF EXPERIMENTALLY OBTAINED FUEL FLOW RATES. JOHN DEERE, MODEL 4239T, WITH AERODYNE TURBOCHARGER, +10° NOZZLE LEVER POSITION

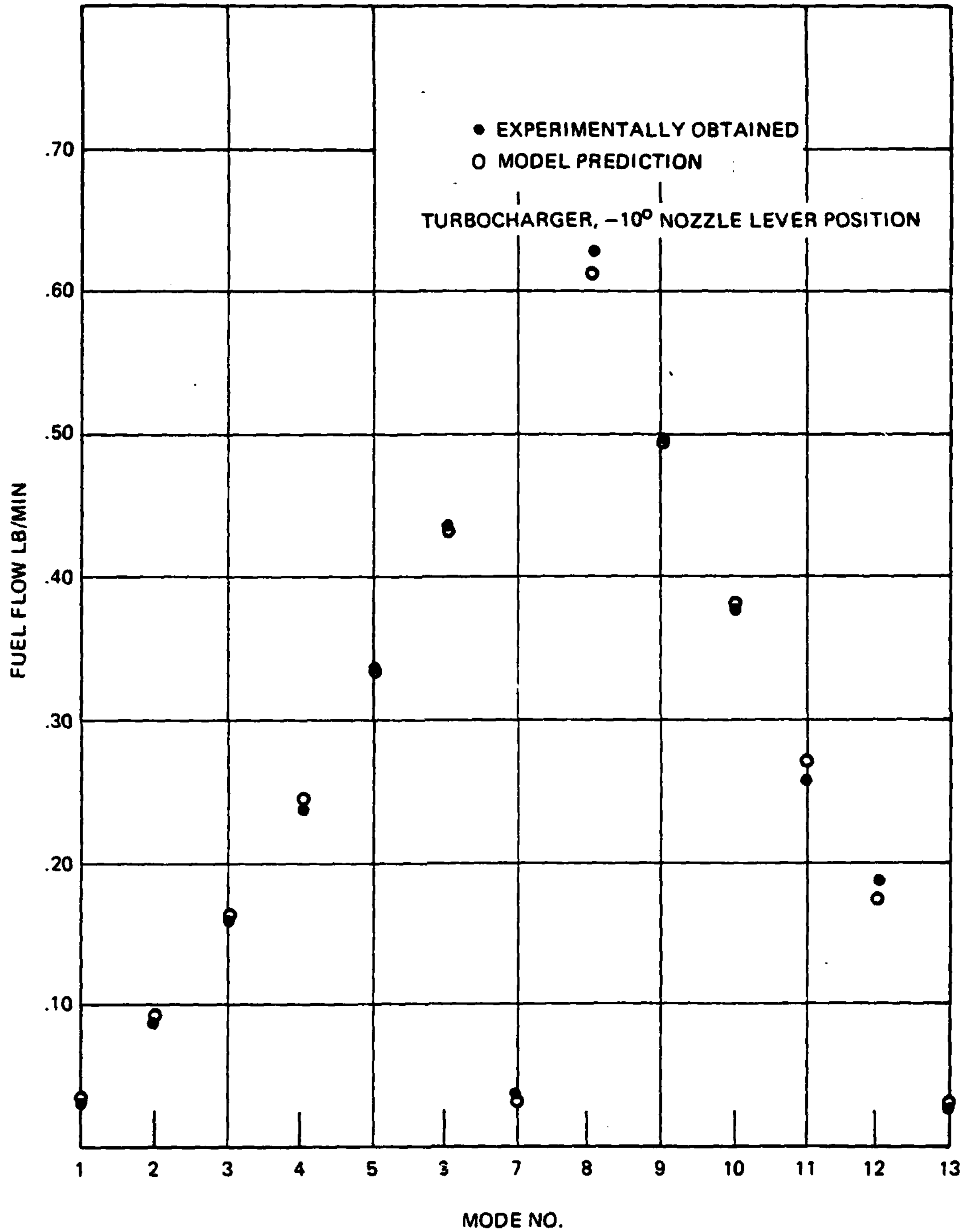


FIGURE 19 - COMPARISON OF MODEL PREDICTIONS WITH THOSE OF EXPERIMENTALLY OBTAINED FUEL FLOW RATES. JOHN DEERE, MODEL 4239T, WITH AERODYNE TURBOCHARGER, -10° NOZZLE LEVER POSITION

N/V RATIOS AND IDLE SPEEDS
EMPLOYED IN THE MODEL

Gear	Case 1*	Case 2	Case 3	Case 4	Case 5
1	205.86	172.48	141.13	109.77	78.405
2	115.76	97.0	76.36	61.72	44.09
3	81.75	68.5	56.0	43.59	31.14
4	65.65	55.0	45.0	35.0	25.0
IDLE SPEED, RPM	975	875	775	675	575

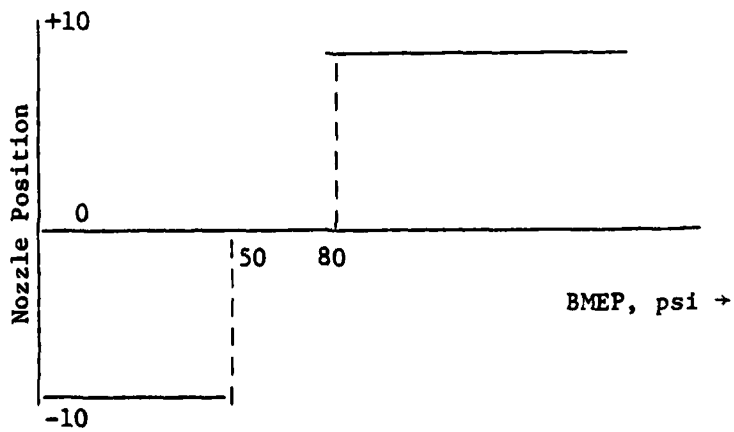
The fuel economy was first estimated over both the cycles for different nozzle positions at a fixed N/V of 35. The results are given below:

MODEL ESTIMATES OF FUEL ECONOMY FOR VARIOUS NOZZLE POSITIONS

N/V in final gear = 35

Nozzle Position	Urban Cycle mpg	Highway Cycle mpg
-10	59	70
0	60	72
+10	61	73
-10°, 0° and 10°	60	72
Best of Three	61	73

In the fourth case of the nozzle position, the following schedule was used to select one of the three nozzle positions.



Schedule of Nozzle Position with BMEP

In the fifth case, called "best of three", the computer estimated the fuel consumption for all the three positions at every second of the cycle (Urban and Highway cycles are respectively 1369 and 764 seconds long) and chose the smallest of the three for estimating the final fuel economy. A counter employed in the program indicates that the fuel consumption with +10° position was the lowest all the time. Therefore, identical results were obtained for both +10° position and "best of three" positions.

The fuel economy for various N/V ratios and the best nozzle position (+10°) were estimated in the same manner and shown below. These results are also plotted with respect to N/V ratio in Figure 20. The fuel economy very nearly increased linearly with decrease in N/V in the range tested.

MODEL ESTIMATES OF FUEL ECONOMY FOR VARIOUS N/V RATIOS

Nozzle Position +10°

N/V in Final Gear	Urban Cycle mpg	Highway Cycle mpg
65.64	45	54
55	50	60
45	56	66
35	61	73
25	65	77

These figures are impressive for a vehicle of 2250 lbs. However, the model was not programmed to check out whether the engine had enough reserve power to accelerate the vehicle over the cycles. The model only predicts the maximum fuel economy theoretically possible by extrapolating the engine and turbocharger characteristics to lower speed operation. Therefore, caution has to be exercised in using these figures. Nevertheless, it is possible to design a low speed engine and produce enough power to drive a 2250 lb. vehicle over these two cycles without difficulty.

Reference 3 (referring to VW test results) indicates that the limit for final N/V is about 38 below which the performance

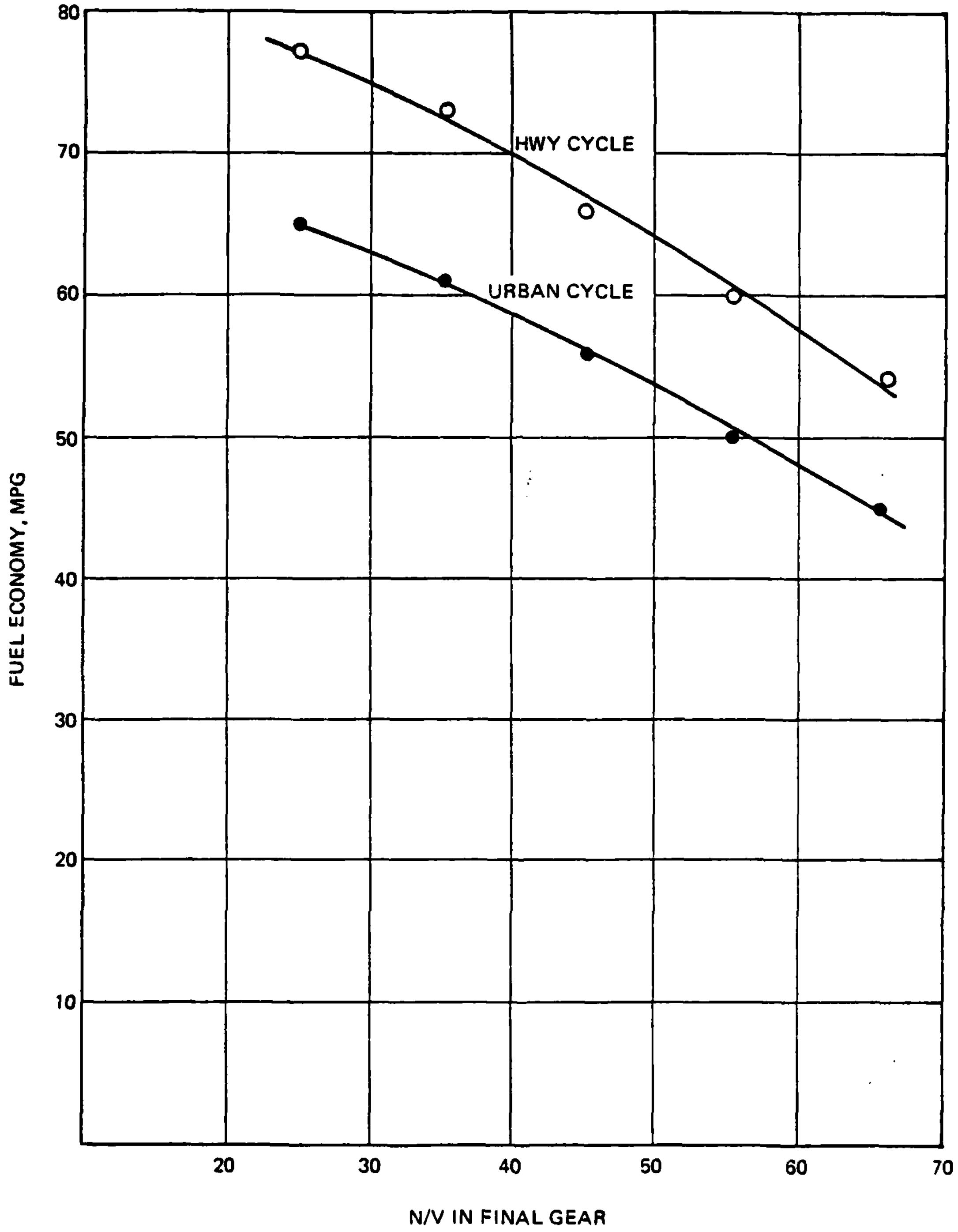


FIGURE 20 - MODEL ESTIMATION OF FUEL FOR VARIOUS N/V RATIOS

(i.e., acceleration) of the turbocharged VW engine in the Rabbit vehicle is not acceptable.

Also note that our model gave higher fuel economy for the naturally aspirated VW engine/vehicle than that experimentally obtained by EPA for the naturally aspirated engine over the EPA Urban cycle. Therefore, all the estimates for the Urban cycle are somewhat high.

VII. CONCLUSIONS

The following conclusions can be drawn from this initial study on a variable nozzle turbocharger:

1. The higher power output obtained from turbocharging allows the engine to operate at lower speeds and obtain higher fuel economy in vehicular applications.

2. Under some steady-state conditions the turbocharger yielded better fuel economy than that obtained with a fixed nozzle turbocharger.

3. At low speeds and part loads, the fuel economy with variable nozzle area turbocharger, set at zero degrees nozzle position, was significantly higher than that with conventional turbocharger, and at higher speeds the differences were not significant. This was probably due to different boost and exhaust pressure characteristics generated by the variable area turbocharger.

4. The variable nozzle area turbocharger did not consistently produce boost pressures higher than the conventional turbocharger at low speeds as expected, probably due to difficulty in reproducing the same nozzle position from test to test.

5. At 2500 rpm and full load conditions, the nozzle position regulated the boost pressure ratio between 1.5 and 2.29. This range of control is probably adequate to eliminate the need for a "waste gate" in the system.

6. On theoretical grounds, it was expected that an engine equipped with a variable nozzle area turbocharger would produce different power outputs with different turbonozzle areas. However, when tested under transient conditions, there was no change in transient response of the engine between zero and +10° nozzle lever positions.

7. The mathematical model predicts that a vehicle similar

to the VW Rabbit equipped with an Aerodyne type variable nozzle turbocharger yields highest fuel economy over EPA driving cycles with +10° nozzle position.

8. The model estimates that the fuel economy increases almost linearly with decreases in final N/V ratio.

9. Either turbocharger used over the standard 13-mode Federal Heavy Duty Engine Emission cycle decreased hydrocarbon and carbon monoxide emissions by about four fold and increased the oxides of nitrogen about two times.

10. Either turbocharger decreased the smoke emission in the entire operating range of the engine. The Aerodyne turbocharger operating at the 10° position was somewhat better in smoke emissions than the production turbocharger at almost all speed-load conditions.

11. In general, the air delivery and compressor efficiencies of the variable nozzle turbocharger were lower than those of the production turbocharger.

REFERENCES

1. Taylor, C. F., "The Internal Combustion Engine in Theory and Practice", Volumes 1 and 2, Second Edition, 1977, the M.I.T. Press, Massachusetts Institute of Technology, Cambridge, Massachusetts.
2. OLert, E. F., "Internal Combustion Engines and Air Pollution", Intext Educational Publishers.
3. Report No. DOT-TSC-NHTSA-77-3, 1, "Data Base for Light-Weight Automotive Diesel Powerplants", Prepared by Volkswagnwerk for U.S. Department of Transportation, National Highway Traffic Safety Administration, Office of Vehicle Systems Research, Washington, DC 20590.

APPENDIX A
CHOICE OF THE ENGINE

CHOICE OF THE ENGINE

The first task in this program was to select a diesel engine to match the turbocharger being developed at Aerodyne Dallas. A map of the compressor characteristics was furnished by Aerodyne for selecting a suitable engine. This map is shown in Figure A-1. The characteristics portrayed in this map were estimates and intended only for guidance.

Engine Size Determination

Normally, a turbocharger is selected from the performance characteristics of an engine. In this case, owing to the special nature of this project, an engine was selected from the characteristics of the turbocharger. The following procedure was followed:

According to Figure A-1, the compressor has a maximum flow capacity of about 320 cubic feet per minute (CFM). When this flow is compressed, the temperature and specific volume of the air in the intake system change. These thermodynamic quantities were determined and are shown in dimensionless form for various pressure boosts and compressor efficiencies in Figure A-2. Note that the plots indicated by $\eta_c = 1.0$ are for reversible adiabatic (isentropic) compression. Also computed and shown in Figure A-3 is the volume rate of air consumption for a 4-stroke engine. For the purpose of selecting the size of the engine, it was assumed that compressor and volumetric efficiencies would be 70% and 80%, respectively. A point shown by a circle on the high side of the compressor map (Figure A-1) was chosen as a design point. This circle marks the point at which the compressor is capable of delivering 285 CFM with 70% efficiency at a pressure boost ratio of 2.9. When the air is compressed to a 2.9 pressure ratio, the specific volume at the outlet decreases to 52% of inlet specific volume (Figure A-2). Hence, the volume rate of flow to the engine would be 148.2 CFM. To achieve this flow rate, the engine, which has a volumetric efficiency of 80%, should have a maximum NV_d (speed x CID) of about 6.4×10^5 (Figure A-3). Diesel engines, however, vary in their maximum speed. Therefore, another plot

(Figure A-4) for determining the displacement of the engine from NV_d was written. For the case of $NV_d = 6.4 \times 10^5$, if the maximum speed is 3500 rpm, the displacement would be about 190 cubic inches. On the other hand, if the maximum speed is only 2500 rpm, the displacement would be about 250 cubic inches.

Additional Considerations

The other factors which were considered include availability of parts, combustion chamber design, and type of intake system. For the purposes of this program in which the characteristics of the variable nozzle turbocharger are to be determined, the design of the combustion chamber (open or precombustion chamber) would not make a great deal of difference. Since it was intended as a part of the testing plan to compare the performance of the variable nozzle turbocharger with that of the fixed nozzle type of turbocharger, an engine already equipped with a turbocharger would be preferable.

A list of available diesel engines in the size range of 175 to 250 cubic inches is included in Table 1. In this size range, there is only one engine which is equipped with a turbocharger. This is the John Deere Model 4239T. Therefore, this engine was chosen. If a higher speed engine was to be selected, the Perkins Model 6-247 would have been a logical choice. The specifications of the Deere Model 4239T are shown in Table 2.

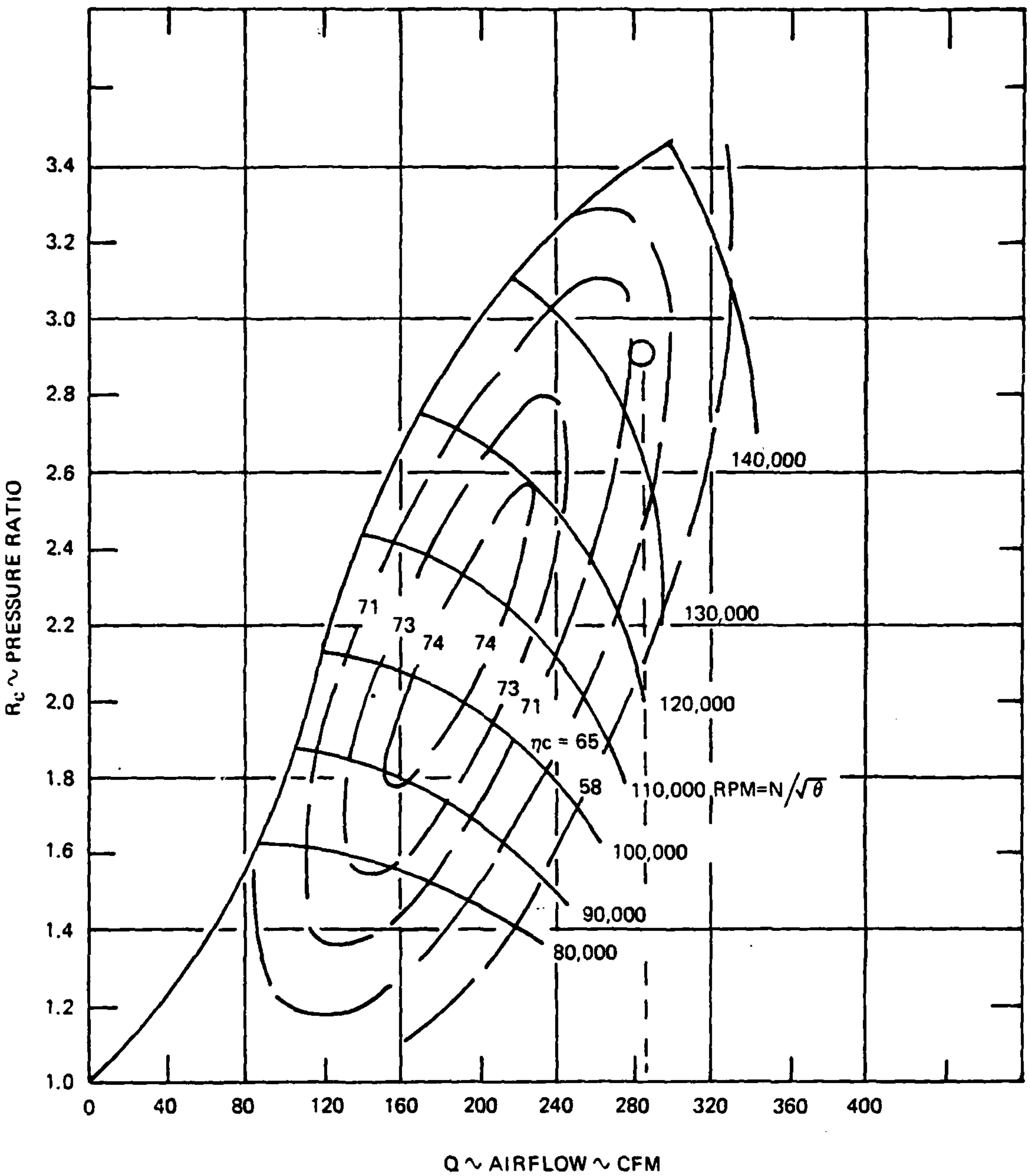


FIGURE A-1 - ESTIMATED MAP OF AERODYNE COMPRESSOR

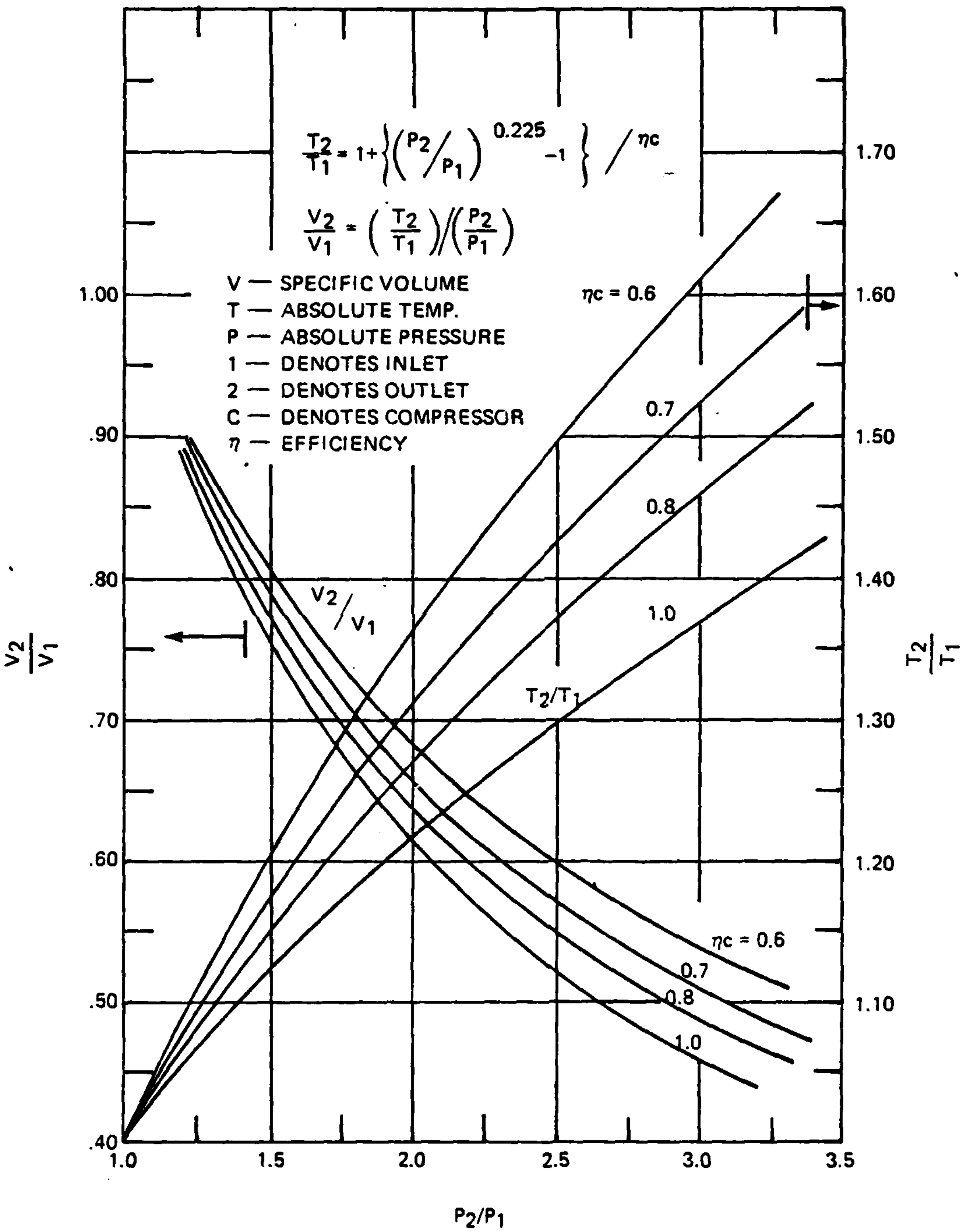


FIGURE A-2 - VARIATION OF V_2/V_1 WITH RESPECT TO PRESSURE BOOST, P_2/P_1

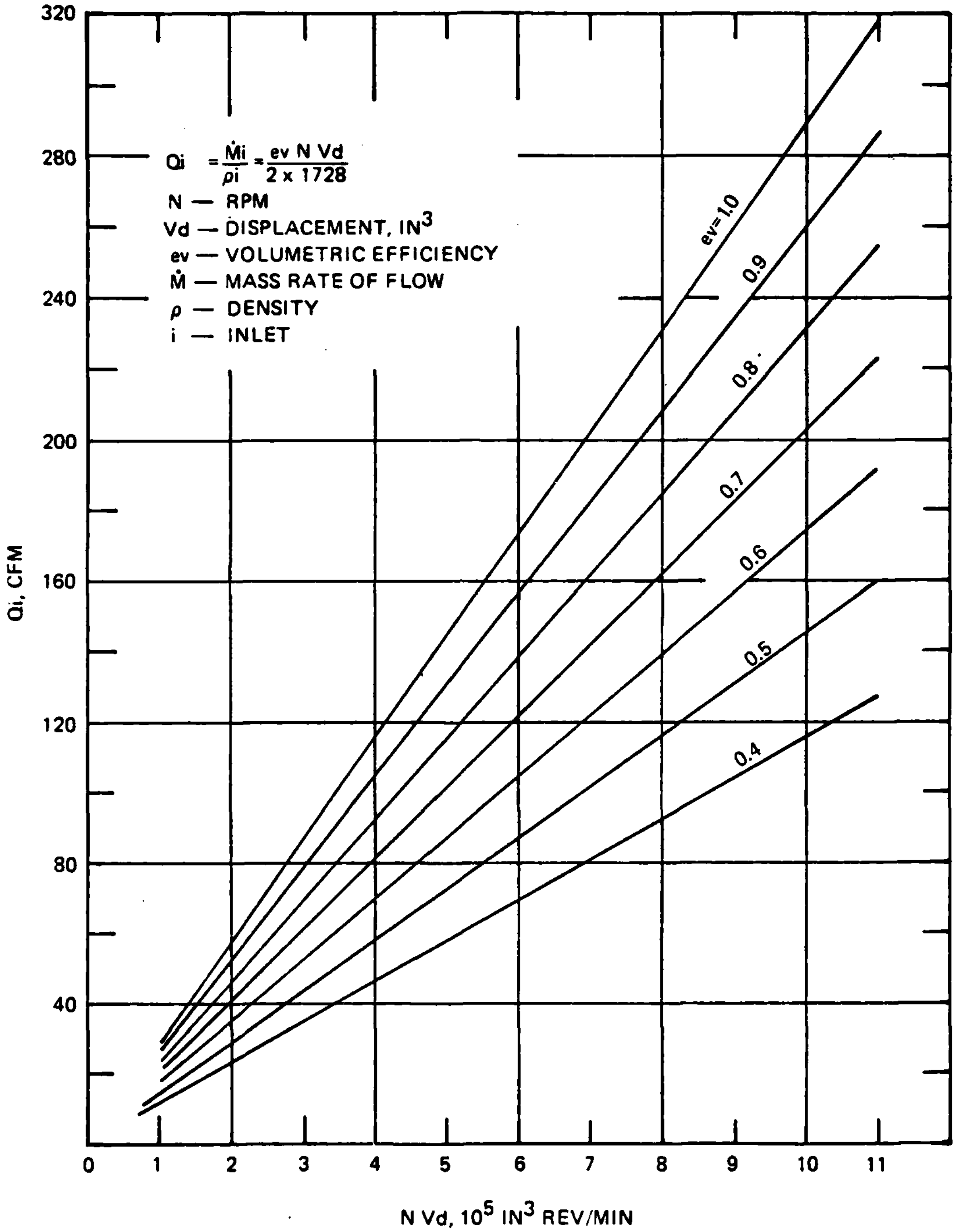


FIGURE A-3 - AIR CONSUMPTION RATE OF A 4-STROKE ENGINE

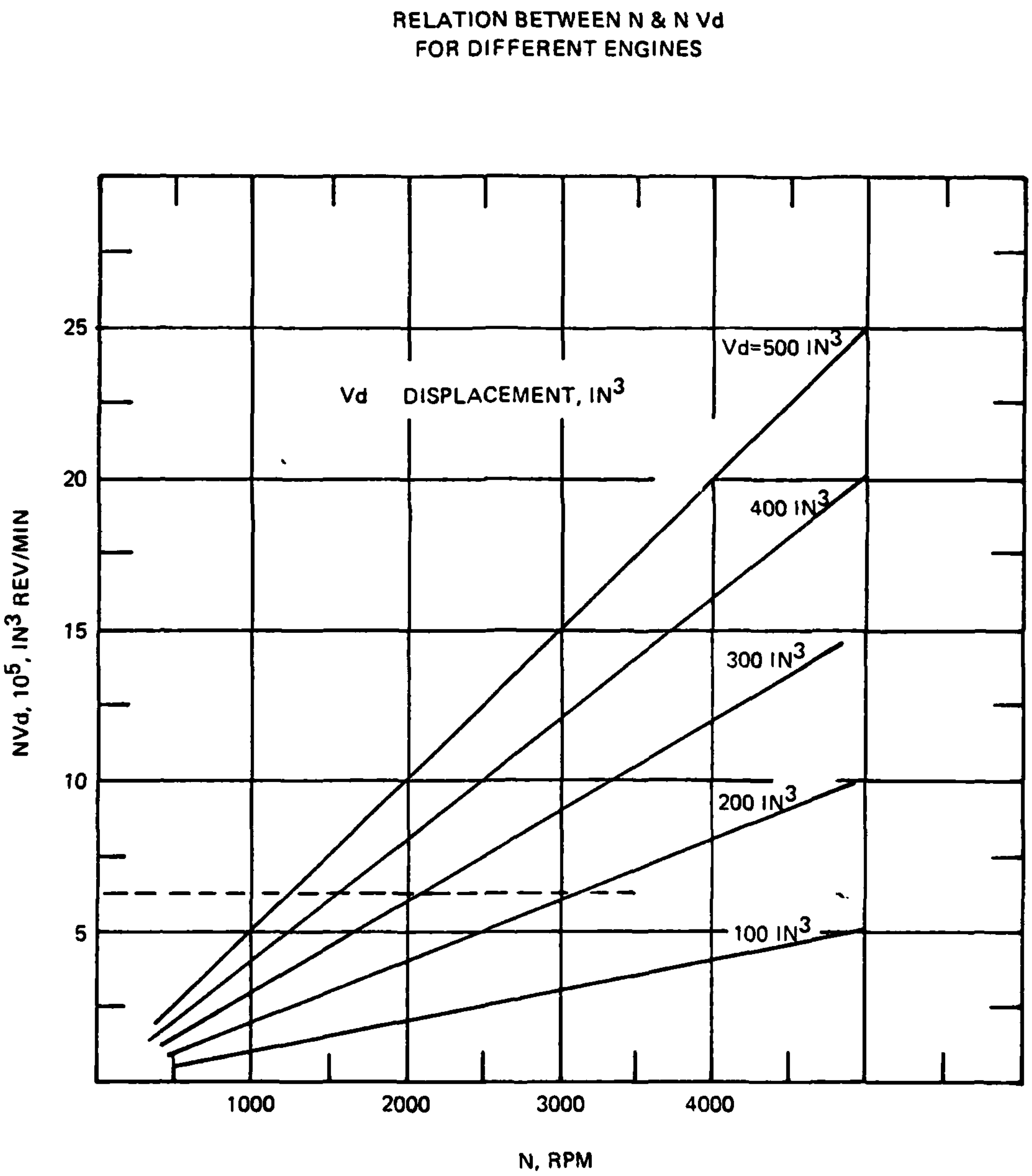


FIGURE A-4 - RELATION BETWEEN N & N Vd FOR DIFFERENT ENGINES

TABLE A-1

List of 4-Stroke Diesel Engines
175 to 250 Cubic Inches

Make	Model	Number of Cylinders and Displacement	Intake System	Maximum Intermittent HP @ RPM
Case	188D	4-188	NA*	55 @ 2200
Chrysler	IN633	6-198	NA	73 @ 3200
Chrysler	C1641	6-243	NA	103 @ 3500
Deere	4219D	4-219	NA	70 @ 2500
Deere	4239T	4-239	T**	89 @ 2500
Ford	201D	3-201	NA	56 @ 2200
Ford	233D	4-233	NA	63 @ 2100
Ford	192DF	4-192	NA	52 @ 2400
Ford	254DF	4-254	NA	70 @ 2500
Perkins	4-203	4-203	NA	54 @ 2400
Perkins	4-236	4-236	NA	77 @ 2500
Perkins	6-247	6-247	NA	-- 3500
Waukesha	VRD232	6-232	NA	-- 2200
White	D 2000	4-198	NA	60 @ 2600
White Farm	2-60	4-211	NA	---------

*NA - naturally-aspirated
** - turbocharged

TABLE A-2

Test Engine Specifications

Engine Make and Model	Deere, 4239T
Number of Cylinders	4
Bore, in.	4.19
Stroke, in.	4.33
Displacement, cu. in.	239
Compression Ratio	16.3
Rated Speed, rpm	2500
HP (Intermittent) @ RS/w/o Fan	89
HP (Continuous) w/o Fan	70 @ 2200
Normal Speed Range, rpm	1500-2500
Low Idle, rpm	800
Torque @ rpm (Max) w/o Fan, ft • lb	208 @ 1700
Basic Weight, lb	950

APPENDIX B

BASELINE TEST DATA AND RESULTS

TABLE B-1

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

NATURALLY-ASPIRATED, 1000 RPM JULY 20, 1978

BARO PR, IN HG	29.11	29.11	29.11	29.11
DRY BULB TEMP, F	80	80	80	80
WET BULB, F	74	74	74	74
ENGINE SPEED, RPM	1000	1000	1000	1000
DYNO LOAD, LB	93.00	69.00	46.00	23.00
POWER OUTPUT, HP	23.25	17.25	11.50	5.75
BMEP, PSI	77.11	57.21	38.14	19.07
AIR FLOW				
LFE DIFF PR, IN H20	.48	.41	.40	.40
LFE PR, IN H20	.10	.10	.10	.10
LFE TEMP, F	82	84	84	84
PCF	.973	.973	.973	.973
TCF	.9611	.9548	.9548	.9548
AIR RATE, LB/HR	238.02	201.98	197.06	197.06
FUEL FLOW				
TIME FOR 1 LB, SEC	316.0	481.2	644.0	944.4
FUEL RATE, LB/HR	11.39	7.48	5.59	3.81
BSFC,LB/HP.HR	.490	.434	.486	.663
TEMPERATURES				
COOLANT IN, F	80	79	79	79
COOLANT OUT, F	178	176	0	176
OIL SUMP, F	195	196	195	190
AMB AIR, F	82	84	84	84
INTAKE MANI, F	90	91	90	90
EXH MANI, F	803	631	520	386
PRESSURES				
INTAKE MANI, IN HG	-.60	-.50	-.40	-.40
EXH MANI, IN HG	.05	.05	.05	.05
AIR-FUEL RATIO	20.89	27.00	35.25	51.69
ENGINE VOL EFF, %	83.6	70.8	68.7	68.7
ENGINE BR TH EFF, %	28.2	31.9	28.4	20.8

STOP

TABLE B-2

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

NATURALLY-ASPIRATED, 1500 RPM JULY 20, 1978

BARO PR, IN HG	29.11	29.11	29.11	29.11
DRY BULB TEMP, F	88	88	88	88
WET BULB, F	75	75	75	75
ENGINE SPEED, RPM	1500	1500	1500	1500
DYNO LOAD, LB	97.00	74.00	49.00	24.00
POWER OUTPUT, HP	36.38	27.75	18.38	9.00
BMEP, PSI	80.42	61.35	40.63	19.90
AIR FLOW				
LFE DIFF PR, IN H20	.64	.64	.64	.66
LFE PR, IN H20	.10	.10	.10	.20
LFE TEMP, F	88	88	89	89
PCF	.973	.973	.973	.972
TCF	.9425	.9425	.9395	.9395
AIR RATE, LB/HR	311.23	311.23	310.23	319.84
FUEL FLOW				
TIME FOR 1 LB, SEC	236.0	300.4	408.0	599.2
FUEL RATE, LB/HR	15.25	11.98	8.82	6.01
BSFC,LB/HP.HR	.419	.432	.480	.668
TEMPERATURES				
COOLANT IN, F	84	83	83	81
COOLANT OUT, F	179	178	176	175
OIL SUMP, F	211	210	208	205
AMB AIR, F	88	88	88	88
INTAKE MANI, F	93	92	92	92
EXH MANI, F	800	728	578	429
PRESSURES				
INTAKE MANI, IN HG	-.60	-.50	-.40	-.40
EXH MANI, IN HG	.10	.10	.10	.10
AIR-FUEL RATIO	20.40	25.97	35.16	53.24
ENGINE VOL EFF, %	73.2	72.9	72.4	74.6
ENGINE BR TH EFF, %	32.9	32.0	28.8	20.7

STOP

TABLE B-3

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

NATURALLY-ASPIRATED, 2000 RPM JULY 20, 1978

BARO PR, IN HG	29.15	29.15	29.15	29.15
DRY BULB TEMP, F	90	90	90	91
WET BULB, F	75	75	75	75
ENGINE SPEED, RPM	2000	2000	2000	2000
DYNO LOAD, LB	92.00	69.00	46.00	23.00
POWER OUTPUT, HP	46.00	34.50	23.00	11.50
BMEP, PSI	76.28	57.21	38.14	19.07
AIR FLOW				
LFE DIFF PR, IN H20	.82	.83	.84	.85
LFE PR, IN H20	.20	.20	.20	.20
LFE TEMP, F	92	92	92	92
PCF	.974	.974	.974	.974
TCF	.9305	.9305	.9305	.9305
AIR RATE, LB/HR	394.11	398.92	403.72	408.53
FUEL FLOW				
TIME FOR 1 LB, SEC	184.4	228.8	302.4	411.2
FUEL RATE, LB/HR	19.52	15.73	11.90	8.75
BSFC,LB/HP.HR	.424	.456	.518	.761
TEMPERATURES				
COOLANT IN, F	84	82	83	81
COOLANT OUT, F	179	178	177	176
OIL SUMP, F	221	219	216	211
AMB AIR, F	92	92	92	92
INTAKE MANI, F	95	95	95	95
EXH MANI, F	918	780	622	501
PRESSURES				
INTAKE MANI, IN HG	-.60	-.50	-.50	-.40
EXH MANI, IN HG	.20	.20	.20	.20
AIR-FUEL RATIO	20.19	25.35	33.91	46.66
ENGINE VOL EFF, %	69.7	70.3	71.2	71.8
ENGINE BR TH EFF, %	32.6	30.3	26.7	18.1

STOP

B-4

TABLE B-4

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

NATURALLY-ASPIRATED, 2500 RPM JULY 20, 1978

BARO PR, IN HG	29.15	29.15	29.15	29.15
DRY BULB TEMP, F	91	91	91	93
WET BULB, F	75	75	75	78
ENGINE SPEED, RPM	2500	2500	2500	2500
DYNO LOAD, LB	76.00	57.00	38.00	19.00
POWER OUTPUT, HP	47.50	35.63	23.75	11.88
BMEP, PSI	63.01	47.26	31.51	15.75
AIR FLOW				
LFE DIFF PR, IN H20	.98	1.00	1.00	1.00
LFE PR, IN H20	.30	.30	.30	.30
LFE TEMP, F	93	95	94	95
PCF	.974	.974	.974	.974
TCF	.9275	.9216	.9246	.9216
AIR RATE, LB/HR	469.39	475.92	477.44	475.92
FUEL FLOW				
TIME FOR 1 LB, SEC	156.4	188.4	226.4	258.4
FUEL RATE, LB/HR	23.02	19.11	15.90	13.93
BSFC,LB/HP.HR	.485	.536	.670	1.173
TEMPERATURES				
COOLANT IN, F	87	85	85	84
COOLANT OUT, F	180	179	177	177
OIL SUMP, F	230	228	225	223
AMB AIR, F	93	95	94	95
INTAKE MANI, F	97	96	96	96
EXH MANI, F	999	872	766	670
PRESSURES				
INTAKE MANI, IN HG	-.70	-.60	-.50	-.40
EXH MANI, IN HG	0.00	0.00	0.00	0.00
AIR-FUEL RATIO	20.39	24.91	30.03	34.16
ENGINE VOL EFF, %	66.9	67.5	67.5	67.0
ENGINE BR TH EFF, %	28.5	25.8	20.6	11.8

STOP

TABLE B-5

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

WITH TURBOCHARGER, 1000 RPM   JULY 19, 1978

BARO PR, IN HG	29.15	29.15	29.15	29.15
DRY BULB TEMP, F	94	94	94	96
WET BULB, F	86	86	86	86
ENGINE SPEED, RPM	1000	1000	1000	1000
DYNO LOAD, LB	124.00	93.00	62.00	31.00
POWER OUTPUT, HP	31.00	23.25	15.50	7.75
BMEP, PSI	102.81	77.11	51.40	25.70
AIR FLOW				
LFE DIFF PR, IN H2O	.81	.62	.50	.46
LFE PR, IN H2O	.20	.20	.10	.10
LFE TEMP, F	96	96	97	97
PCF	.974	.974	.974	.974
TCF	.9187	.9187	.9158	.9158
AIR RATE, LB/HR	384.37	294.21	236.57	217.65
FUEL FLOW				
TIME FOR 1 LB, SEC	186.4	290.0	458.0	768.4
FUEL RATE, LB/HR	19.31	12.41	7.86	4.69
BSFC,LB/HP.HR	.623	.534	.507	.605
TEMPERATURES				
COOLANT IN, F	86	84	82	81
COOLANT OUT, F	179	178	177	176
OIL SUMP, F	222	216	209	199
AMB AIR, F	96	96	97	97
COMP INLET, F	97	97	99	99
COMP OUTLET, F	168	139	125	116
TURBO INLET, F	959	775	670	437
TURBO OUTLET, F	860	701	564	410
PRESSURES				
COMP INLET, IN H2O	2.15	1.70	.95	.70
COMP OUTLET, IN HG	8.40	4.10	1.50	.35
TURBO INLET, IN HG	6.30	3.70	2.30	1.70
TURBO OUTLET, IN HG	.10	.05	.05	.05
AIR-FUEL RATIO	19.90	23.70	30.10	46.46
ENGINE VOL EFF, %	117.0	96.5	82.2	77.3
ENGINE BR TH EFF, %	22.2	25.9	27.2	22.9
COMP PR BOOST	1.30	1.15	1.05	1.01
VALUE OF YC	.077	.039	.015	.004
COMP TEMP DIFF, F	71	42	26	17
COMP ISENTROPIC EFF, %	60.0	52.4	32.5	12.9
Pc/pt	.914	.988	1.026	1.016

STOP

TABLE B-6

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

WITH TURBOCHARGER, 1500 RPM JULY 18, 1978

BARO PR, IN HG	29.10	29.10	29.10	29.10
DRY BULB TEMP, F	82	82	82	95
WET BULB, F	73	73	73	73
ENGINE SPEED, RPM	1500	1500	1500	1500
DYNO LOAD, LB	106.00	79.50	53.00	26.50
POWER OUTPUT, HP	39.75	29.81	19.88	9.94
BMEP, PSI	87.88	65.91	43.94	21.97
AIR FLOW				
LFE DIFF PR, IN H20	.76	.74	.70	.68
LFE PR, IN H20	.20	.20	.20	.20
LFE TEMP, F	101	101	99	99
PCF	.972	.972	.972	.972
TCF	.9043	.9043	.9100	.9100
AIR RATE, LB/HR	354.38	345.05	328.46	319.08
FUEL FLOW				
TIME FOR 1 LB, SEC	220.8	270.4	316.4	603.6
FUEL RATE, LB/HR	16.30	13.31	11.38	5.96
BSFC,LB/HP.HR	.410	.447	.572	.600
TEMPERATURES				
COOLANT IN, F	84	82	82	80
COOLANT OUT, F	179	179	177	176
OIL SUMP, F	221	213	208	203
AMB AIR, F	101	101	99	99
COMP INLET, F	103	101	100	100
COMP OUTLET, F	159	145	132	122
TURBO INLET, F	880	752	595	425
TURBO OUTLET, F	798	685	555	399
PRESSURES				
COMP INLET, IN H20	2.10	2.00	1.90	1.85
COMP OUTLET, IN HG	6.40	4.90	2.90	1.60
TURBO INLET, IN HG	5.40	4.80	3.95	3.30
TURBO OUTLET, IN HG	.05	.05	.02	.02
AIR-FUEL RATIO	21.74	25.92	28.87	53.50
ENGINE VOL EFF, %	75.0	74.5	73.7	73.4
ENGINE BR TH EFF, %	33.7	30.9	24.1	23.0
COMP PR BOOST	1.23	1.17	1.10	1.06
VALUE OF YC	.060	.047	.029	.017
COMP TEMP DIFF, F	56	44	32	22
COMP ISENTROPIC EFF, %	60.2	59.7	50.5	42.6

STOP

TABLE B-7

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

WITH TURBOCHARGER, 2000 RPM JULY 19, 1978

BARO PR, IN HG	29.15	29.15	29.15	29.15
DRY BULB TEMP, F	82	82	82	83
WET BULB, F	76	76	76	75
ENGINE SPEED, RPM	2000	2000	2000	2000
DYNO LOAD, LB	140.00	105.00	70.50	35.25
POWER OUTPUT, HP	70.00	52.50	35.25	17.63
BMEP, PSI	116.07	87.06	58.45	29.23
AIR FLOW				
LFE DIFF PR, IN H2O	1.18	1.08	1.02	.94
LFE PR, IN H2O	.40	.30	.30	.20
LFE TEMP, F	85	85	85	85
PCF	.973	.974	.974	.974
TCF	.9517	.9517	.9517	.9517
AIR RATE, LB/HR	579.79	530.79	501.30	462.10
FUEL FLOW				
TIME FOR 1 LB, SEC	126.8	161.2	230.8	338.8
FUEL RATE, LB/HR	28.39	22.33	15.60	10.63
BSFC,LB/HP.HR	.406	.425	.442	.603
TEMPERATURES				
COOLANT IN, F	88	86	84	84
COOLANT OUT, F	180	179	178	177
OIL SUMP, F	234	231	221	216
AMB AIR, F	85	85	85	85
COMP INLET, F	85	86	86	86
COMP OUTLET, F	208	182	162	133
TURBO INLET, F	1001	864	777	547
TURBO OUTLET, F	892	779	693	507
PRESSURES				
COMP INLET, IN H2O	3.60	3.25	2.90	2.65
COMP OUTLET, IN HG	16.70	12.20	8.20	4.80
TURBO INLET, IN HG	12.80	10.80	8.70	7.00
TURBO OUTLET, IN HG	.15	.12	.10	.05
AIR-FUEL RATIO	20.42	23.77	32.14	43.49
ENGINE VOL EFF, %	76.9	75.0	76.0	73.5
ENGINE BR TH EFF, %	34.1	32.5	31.2	22.9
COMP PR BOOST	1.59	1.43	1.29	1.17
VALUE OF YC	.141	.107	.075	.046
COMP TEMP DIFF, F	123	96	76	47
COMP ISENTROPIC EFF, %	62.4	61.1	54.2	53.9
VALUE OF YT	.088	.077	.064	.053
TURB EFF, %	81.3	80.8	82.9	76.2

STOP

TABLE B-8

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

WITH TURBOCHARGER, 2500 RPM JULY 19, 1978

BARO PR, IN HG	29.15	29.15	29.15	29.15
DRY BULB TEMP, F	83	83	83	83
WET BULB, F	75	75	75	75
ENGINE SPEED, RPM	2500	2500	2500	2500
DYNO LOAD, LB	148.00	111.00	73.50	37.00
POWER OUTPUT, HP	92.50	69.38	45.94	23.13
BMEP, PSI	122.71	92.03	60.94	30.68
AIR FLOW				
LFE DIFF PR, IN H20	1.69	1.54	1.44	1.30
LFE PR, IN H20	.60	.60	.40	.40
LFE TEMP, F	86	87	88	88
PCF	.973	.973	.973	.973
TCF	.9487	.9456	.9425	.9425
AIR RATE, LB/HR	827.27	751.41	700.70	632.58
FUEL FLOW				
TIME FOR 1 LB, SEC	91.2	115.6	146.4	202.8
FUEL RATE, LB/HR	39.47	31.14	24.59	17.75
BSFC,LB/HP.HR	.427	.449	.535	.768
TEMPERATURES				
COOLANT IN, F	94	92	88	85
COOLANT OUT, F	182	180	179	179
OIL SUMP, F	239	242	233	228
AMB AIR, F	85	87	88	88
COMP INLET, F	88	88	89	88
COMP OUTLET, F	286	246	216	173
TURBO INLET, F	1105	985	875	685
TURBO OUTLET, F	933	847	764	616
PRESSURES				
COMP INLET, IN H20	6.80	5.80	5.30	4.25
COMP OUTLET, IN HG	30.40	23.00	17.50	11.30
TURBO INLET, IN HG	23.30	19.50	16.70	12.80
TURBO OUTLET, IN HG	.40	.35	.20	.15
AIR-FUEL RATIO	20.96	24.13	28.50	35.64
ENGINE VOL EFF, %	75.4	74.1	73.9	72.1
ENGINE BR TH EFF, %	32.4	30.8	25.8	18.0
COMP PR BOOST	2.08	1.82	1.62	1.40
VALUE OF YC	.232	.185	.148	.101
COMP TEMP DIFF, F	198	158	127	85
COMP ISENTROPIC EFF, %	64.2	64.3	63.9	65.3
VALUE OF YT	.137	.120	.108	.088
TURB EFF, %	78.6	77.6	75.7	73.1

STOP

TABLE B-9

Diesel Fuel Properties

Gravity, API No.	35.4
Specific Gravity	0.8478 @ 60°F
Percent Weight of Carbon	86.11
Percent Weight of Hydrogen	13.16
Higher Heating Value, BTU/lb	18422
Lower Heating Value, BTU/lb	17622
Hydrocarbon Composition	
Saturates, % V	70.2
Aromatics, % V	28.6
Olefins, % V	1.2

APPENDIX C

MAXIMUM POWER OUTPUT TEST RESULTS
VARIABLE NOZZLE AREA TC

TABLE C-1

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger JAN 4, 79

BARO PR, IN HG	29.66	29.66	29.66	29.66
DRY BULB TEMP, F	62	62	62	62
WET BULB, F	55	55	55	55
ENGINE SPEED, RPM	1000	1500	2000	2500
DYNO LOAD, LB	171.00	182.00	183.00	156.00
POWER OUTPUT, HP	42.75	68.25	91.50	97.50
BMEP, PSI	141.78	150.90	151.73	129.34
TURBO NOZZLE POS, DEG	0.0	0.0	0.0	0.0
TURBO ROTOR SPEED, RPM	61400	70900	92800	106700
AIR FLOW				
LFE DIFF PR, IN H2O	.48	.74	1.16	1.48
LFE PR, IN H2O	.20	.30	.50	.60
LFE TEMP, F	70	69	68	67
PCF	.991	.991	.990	.990
TCF	1.0000	1.0034	1.0068	1.0101
AIR RATE, LB/HR	252.28	390.13	613.31	784.95
FUEL FLOW				
TIME FOR 1 LB, SEC	184.9	137.2	107.1	91.6
FUEL RATE, LB/HR	19.47	26.24	33.61	39.30
BSFC,LB/HP.HR	.455	.384	.367	.403
TEMPERATURES				
COOLANT IN, F	84	84	83	93
COOLANT OUT, F	182	180	179	180
OIL SUMP, F	220	225	232	226
AMB AIR, F	70	69	68	67
COMP INLET, F	67	66	67	67
COMP OUTLET, F	180	184	227	285
TURBO INLET, F	1185	1220	1118	1114
TURBO OUTLET, F	1127	1100	988	952
PRESSURES				
COMP INLET, IN H2O	.90	2.10	3.90	5.50
COMP OUTLET, IN HG	9.00	13.90	23.80	26.50
TURBO INLET, IN HG	4.10	7.20	14.50	20.20
TURBO OUTLET, IN HG	.05	.20	.30	.50
AIR-FUEL RATIO	12.96	14.87	18.25	19.97
ENGINE VOL EFF, %	76.0	70.0	71.7	75.8
ENGINE BR TH EFF, %	30.3	35.9	37.6	34.3
EXH-INTAKE PR RATIO	.873	.848	.826	.808
COMP PR BOOST	1.31	1.48	1.82	1.92
VALUE OF YC	.079	.117	.186	.204
COMP TEMP DIFF, F	113	118	160	218
TURBO TEMP DIFF, F	58	120	130	162
COMP ISENTROPIC EFF, %	36.9	52.2	61.3	49.4

STOP

MPU= 5.490

TABLE C-2

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger JAN 4, 1979

BARO PR, IN HG	29.66	29.66	29.66	29.66
DRY BULB TEMP, F	62	62	62	62
WET BULB, F	55	55	55	55
ENGINE SPEED, RPM	1000	1500	2000	2500
DYNO LOAD, LB	178.00	192.00	173.00	147.00
POWER OUTPUT, HP	44.50	72.00	86.50	91.88
BMEP, PSI	147.58	159.19	143.43	121.88
TURBO NOZZLE POS, DEG	10.0	10.0	10.0	10.0
TURBO ROTOR SPEED, RPM	71100	91600	115500	127000
AIR FLOW				
LFE DIFF PR, IN H20	.53	.94	1.46	1.66
LFE PR, IN H20	.30	.40	.60	.70
LFE TEMP, F	73	74	73	71
PCF	.991	.990	.990	.990
TCF	.9901	.9868	.9901	.9967
AIR RATE, LB/HR	275.71	487.25	758.94	868.46
FUEL FLOW				
TIME FOR 1 LB, SEC	183.2	140.1	111.2	98.4
FUEL RATE, LB/HR	19.65	25.70	32.37	36.59
BSFC,LB/HP.HR	.442	.357	.374	.398
TEMPERATURES				
COOLANT IN, F	78	84	86	85
COOLANT OUT, F	181	181	181	181
OIL SUMP, F	224	229	237	219
AMB AIR, F	73	74	73	71
COMP INLET, F	70	68	68	70
COMP OUTLET, F	206	239	305	334
TURBO INLET, F	1175	1057	1009	1060
TURBO OUTLET, F	1049	903	794	785
PRESSURES				
COMP INLET, IN H20	1.10	3.00	5.40	6.70
COMP OUTLET, IN HG	13.50	26.50	37.04	37.04
TURBO INLET, IN HG	8.00	16.60	32.00	37.04
TURBO OUTLET, IN HG	.10	.25	.55	.70
AIR-FUEL RATIO	14.03	18.96	23.44	23.74
ENGINE VOL EFF, %	77.4	73.6	79.2	75.3
ENGINE BR TH EFF, %	31.3	38.7	36.9	34.7
EXH-INTAKE PR RATIO	.873	.824	.924	1.000
COMP PR BOOST	1.46	1.91	2.28	2.29
VALUE OF YC	.114	.202	.265	.266
COMP TEMP DIFF, F	136	171	237	264
TURBO TEMP DIFF, F	126	154	215	275
COMP ISENTROPIC EFF, %	44.3	62.4	59.0	53.4

STOP

CPU= 5.588

TABLE C-3

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

JAN 18, 1979

BARO PR, IN HG	29.18	29.18	29.18	29.18
DRY BULB TEMP, F	75	75	75	75
WET BULB, F	68	68	68	68
ENGINE SPEED, RPM	750	750	750	750
DYNO LOAD, LB	179.00	134.25	144.00	108.00
POWER OUTPUT, HP	33.56	25.17	27.00	20.25
BMEP, PSI	148.41	111.31	119.39	89.54
TURBO NOZZLE POS, DEG	10.0	10.0	0.0	0.0
TURBO ROTOR SPEED, RPM	76860	58200	49410	34670
AIR FLOW				
LFE DIFF PR, IN H2O	.62	.46	.40	.33
LFE PR, IN H2O	.20	.20	.20	.10
LFE TEMP, F	75	75	75	75
PCF	.975	.975	.975	.975
TCF	.9835	.9835	.9835	.9835
AIR RATE, LB/HR	315.28	233.92	203.41	167.85
FUEL FLOW				
TIME FOR 1 LB, SEC	161.6	226.8	196.0	328.8
FUEL RATE, LB/HR	22.28	15.87	18.37	10.95
BSFC,LB/HP.HR	.664	.631	.680	.541
TEMPERATURES				
COOLANT IN, F	75	75	75	75
COOLANT OUT, F	179	180	181	180
OIL SUMP, F	173	211	214	213
AMB AIR, F	75	75	76	76
COMP INLET, F	75	75	76	76
COMP OUTLET, F	210	179	175	152
TURBO INLET, F	1170	1050	1170	942
TURBO OUTLET, F	1021	943	1115	849
PRESSURES				
COMP INLET, IN H2O	1.50	.65	.55	.40
COMP OUTLET, IN HG	15.00	7.70	5.10	2.10
TURBO INLET, IN HG	8.50	4.80	2.60	1.50
TURBO OUTLET, IN HG	.15	.05	.05	.02
AIR-FUEL RATIO	14.15	14.74	11.07	15.33
ENGINE VOL EFF, %	116.0	98.4	91.4	79.7
ENGINE BR TH EFF, %	20.8	21.9	20.3	25.6
EXH-INTAKE PR RATIO	.853	.921	.927	.981
COMP PR BOOST	1.52	1.27	1.18	1.07
VALUE OF YC	.127	.070	.047	.020
COMP TEMP DIFF, F	135	104	99	76
TURBO TEMP DIFF, F	149	107	55	93
COMP ISENTROPIC EFF, %	50.2	35.8	25.7	14.3

STOP

MRU= 5.468

::

APPENDIX D

PART LOAD TEST RESULTS
VARIABLE NOZZLE AREA TC

TABLE D-1

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger JAN 4, 1979

BARO PR, IN HG	29.61	29.61	29.61	29.61
DRY BULB TEMP, F	62	62	62	62
WET BULB, F	55	55	55	55
ENGINE SPEED, RPM	1000	1000	1000	1000
DYNO LOAD, LB	31.00	31.00	31.00	31.00
POWER OUTPUT, HP	7.75	7.75	7.75	7.75
BMEP, PSI	25.70	25.70	25.70	25.70
TURBO NOZZLE POS, DEG	0.0	8.0	-8.0	0.0
TURBO ROTOR SPEED, RPM	23570	29390	19600	23620
AIR FLOW				
LFE DIFF PR, IN H2O	.38	.40	.36	.38
LFE PR, IN H2O	.10	.15	.10	.10
LFE TEMP, F	69	68	67	67
PCF	.989	.989	.989	.989
TCF	1.0034	1.0068	1.0101	1.0101
AIR RATE, LB/HR	200.10	211.31	190.85	201.45
FUEL FLOW				
TIME FOR 1 LB, SEC	799.6	806.0	813.2	809.6
FUEL RATE, LB/HR	4.50	4.47	4.43	4.45
BSFC,LB/HP.HR	.581	.576	.571	.574
TEMPERATURES				
COOLANT IN, F	73	90	72	73
COOLANT OUT, F	155	173	174	175
OIL SUMP, F	165	175	181	184
AMB AIR, F	69	68	67	67
COMP INLET, F	68	66	66	66
COMP OUTLET, F	93	96	92	93
TURBO INLET, F	408	407	419	415
TURBO OUTLET, F	356	359	379	375
PRESSURES				
COMP INLET, IN H2O	.80	.80	.75	.80
COMP OUTLET, IN HG	-.10	1.00	-.80	-.10
TURBO INLET, IN HG	1.20	2.30	.70	1.20
TURBO OUTLET, IN HG	.02	.02	.02	.02
AIR-FUEL RATIO	44.44	47.31	43.11	45.30
ENGINE VOL EFF, %	68.2	69.9	66.6	68.7
ENGINE BR TH EFF, %	23.8	24.0	24.2	24.1
EXH-INTAKE PR RATIO	1.044	1.042	1.052	1.044
COMP PR BOOST	1.00	1.04	.97	1.00
VALUE OF YC	-.000	.010	-.007	-.000
COMP TEMP DIFF, F	25	30	26	27
TURBO TEMP DIFF, F	52	48	40	40
COMP ISENTROPIC EFF, %	-.8	17.7	-14.7	-.8

STOP

NRU= 5.450

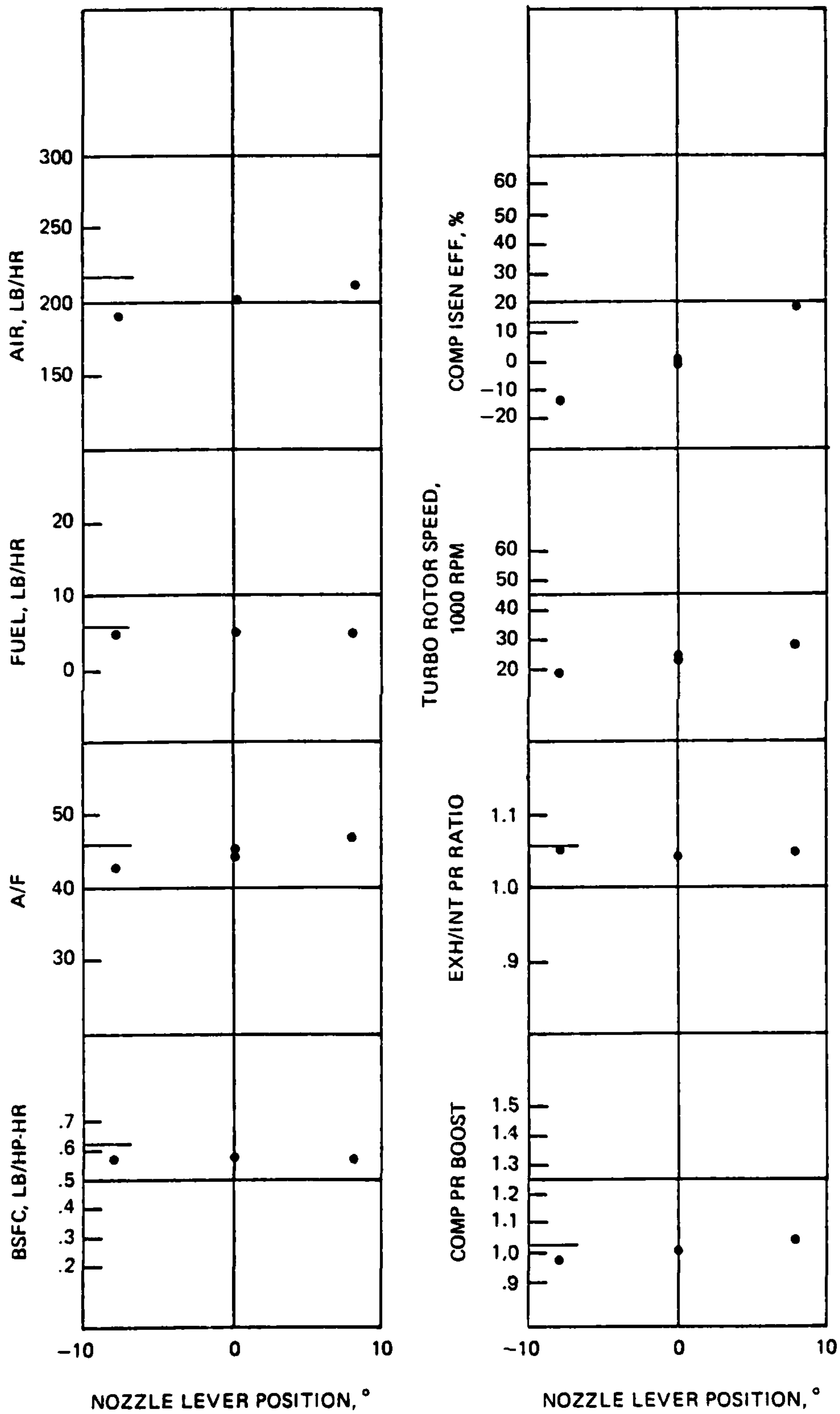


FIGURE D-1 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 1000 RPM and 31 LB LOAD

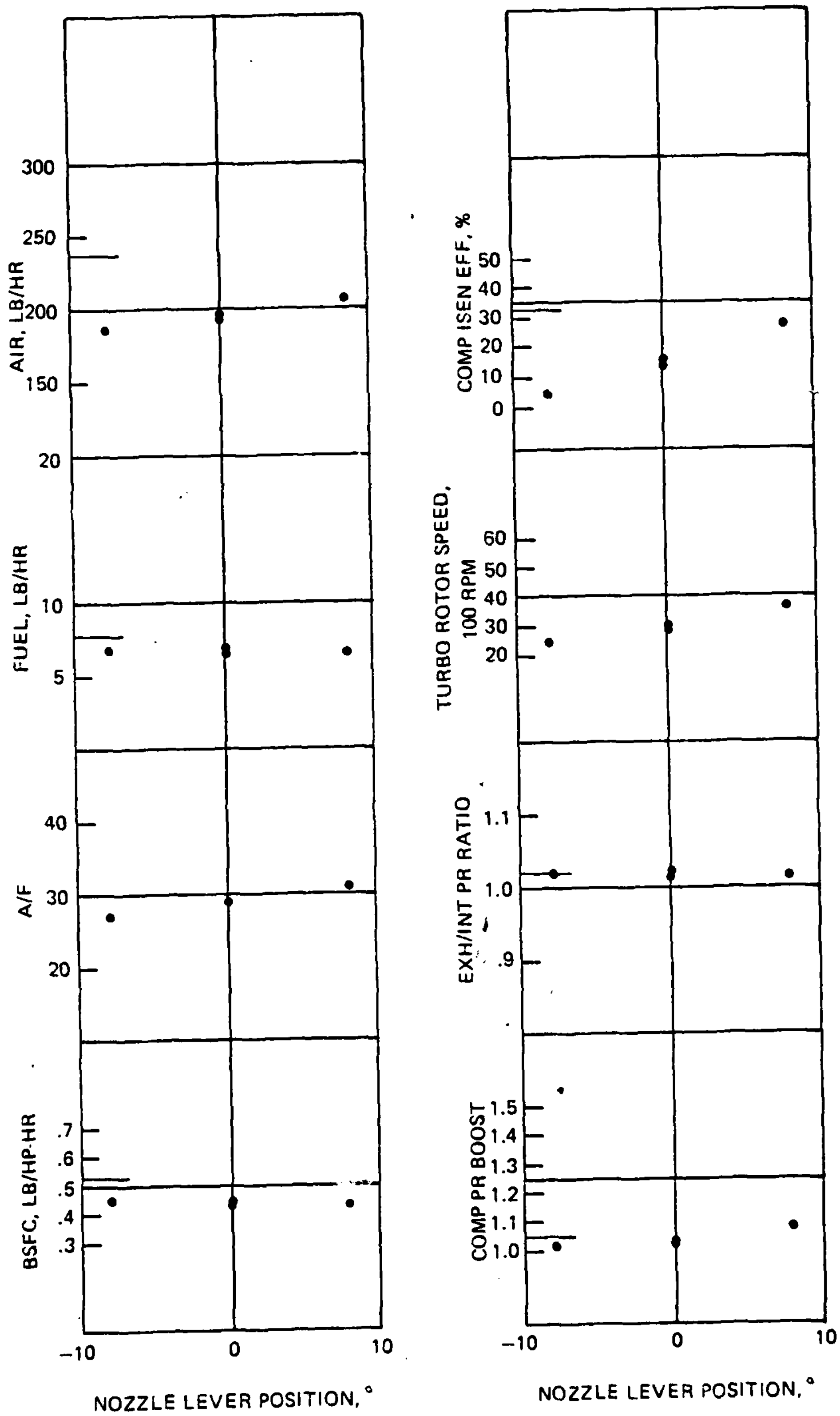


FIGURE D-2 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 1000 RPM AND 61 LB LOAD

TABLE D-2

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger JAN 4, 1979

BARO PR, IN HG	29.54	29.54	29.54	29.54
DRY BULB TEMP, F	62	62	62	62
WET BULB, F	55	55	55	55
ENGINE SPEED, RPM	1000	1000	1000	1000
DYNO LOAD, LB	61.00	61.00	61.00	61.00
POWER OUTPUT, HP	15.25	15.25	15.25	15.25
BMEP, PSI	50.58	50.58	50.58	50.58
TURBO NOZZLE POS, DEG	0.0	8.0	-8.0	0.0
TURBO ROTOR SPEED, RPM	29500	35930	24800	29910
AIR FLOW				
LFE DIFF PR, IN H2O	.38	.40	.36	.38
LFE PR, IN H2O	.15	.20	.15	.15
LFE TEMP, F	70	70	70	71
PCF	.987	.987	.987	.987
TCF	1.0000	1.0000	1.0000	.9967
AIR RATE, LB/HR	198.93	209.38	188.46	198.27
FUEL FLOW				
TIME FOR 1 LB, SEC	525.2	540.8	524.8	527.2
FUEL RATE, LB/HR	6.85	6.66	6.86	6.83
BSFC,LB/HP.HR	.449	.437	.450	.448
TEMPERATURES				
COOLANT IN, F	68	73	73	72
COOLANT OUT, F	177	178	178	178
OIL SUMP, F	182	188	188	193
AMB AIR, F	70	70	70	70
COMP INLET, F	68	69	69	69
COMP OUTLET, F	107	112	106	109
TURBO INLET, F	522	564	590	589
TURBO OUTLET, F	495	486	515	513
PRESSURES				
COMP INLET, IN H2O	.80	.80	.75	.75
COMP OUTLET, IN HG	1.10	2.30	.30	1.10
TURBO INLET, IN HG	1.70	2.80	1.00	1.70
TURBO OUTLET, IN HG	.02	.02	.01	.02
AIR-FUEL RATIO	29.02	31.45	27.47	29.04
ENGINE VOL EFF, %	67.0	68.5	65.1	67.0
ENGINE BR TH EFF, %	30.7	31.6	30.7	30.9
EXH-INTAKE PR RATIO	1.020	1.016	1.023	1.020
COMP PR BOOST	1.04	1.08	1.01	1.04
VALUE OF YC	.011	.022	.003	.011
COMP TEMP DIFF, F	39	43	37	40
TURBO TEMP DIFF, F	27	78	75	76
COMP ISENTROPIC EFF, %	15.0	27.3	4.9	14.6

STOP

?U= 5.666

TABLE D-3

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger JAN 4, 1979

BARO PR, IN HG	29.54	29.54	29.54	29.54
DRY BULB TEMP, F	66	66	66	66
WET BULB, F	55	55	55	55
ENGINE SPEED, RPM	1000	1000	1000	1000
DYNO LOAD, LB	93.00	93.00	93.00	93.00
POWER OUTPUT, HP	23.25	23.25	23.25	23.25
BMEP, PSI	77.11	77.11	77.11	77.11
TURBO NOZZLE POS, DEG	0.0	8.0	-8.0	0.0
TURBO ROTOR SPEED, RPM	38210	45180	32470	38640
AIR FLOW				
LFE DIFF PR, IN H20	.40	.43	.38	.40
LFE PR, IN H20	.15	.20	.15	.15
LFE TEMP, F	71	71	71	72
PCF	.987	.987	.987	.987
TCF	.9967	.9967	.9967	.9934
AIR RATE, LB/HR	208.71	224.33	198.27	208.01
FUEL FLOW				
TIME FOR 1 LB, SEC	361.6	365.6	357.6	360.0
FUEL RATE, LB/HR	9.96	9.85	10.07	10.00
BSFC,LB/HP.HR	.428	.424	.433	.430
TEMPERATURES				
COOLANT IN, F	69	71	69	68
COOLANT OUT, F	179	179	180	179
OIL SUMP, F	197	199	201	202
AMB AIR, F	71	71	71	72
COMP INLET, F	70	70	71	71
COMP OUTLET, F	123	133	122	127
TURBO INLET, F	754	738	789	767
TURBO OUTLET, F	651	639	691	674
PRESSURES				
COMP INLET, IN H20	.80	.80	.80	.80
COMP OUTLET, IN HG	2.80	4.50	1.60	2.90
TURBO INLET, IN HG	2.25	3.80	1.60	2.30
TURBO OUTLET, IN HG	.05	.05	.02	.05
AIR-FUEL RATIO	20.96	22.78	19.69	20.80
ENGINE VOL EFF, %	68.5	71.1	67.4	68.5
ENGINE BR TH EFF, %	32.3	32.6	31.9	32.1
EXH-INTAKE PR RATIO	.983	.979	1.000	.982
COMP PR BOOST	1.10	1.15	1.06	1.10
VALUE OF YC	.027	.042	.016	.028
COMP TEMP DIFF, F	53	63	51	56
TURBO TEMP DIFF, F	103	99	98	93
COMP ISENTROPIC EFF, %	26.7	35.2	16.4	28.2

STOP

CPU= 5.599

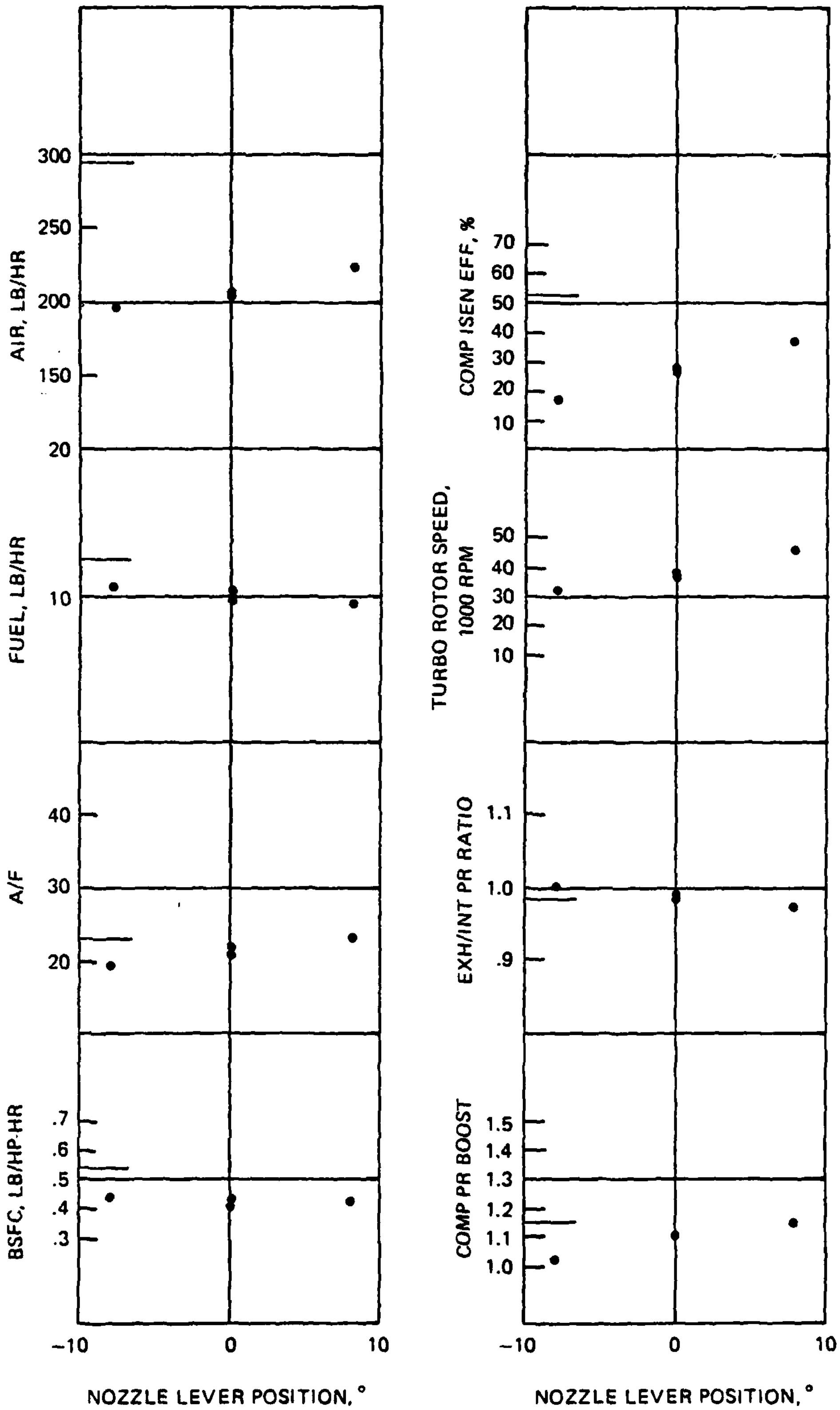


FIGURE D-3 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 1000 RPM AND 93 LB LOAD

TABLE D-4

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Acrodyne Turbocharger JAN 4,79

BARO PR, IN HG	29.54	29.54	29.54	29.54
DRY BULB TEMP, F	62	62	62	62
WET BULB, F	55	55	55	55
ENGINE SPEED, RPM	1000	1000	1000	1000
DYNO LOAD, LB	124.00	124.00	124.00	124.00
POWER OUTPUT, HP	31.00	31.00	31.00	31.00
BMEP, PSI	102.81	102.81	102.81	102.81
TURBO NOZZLE POS, DEG	0.0	8.0	-8.0	0.0
TURBO ROTOR SPEED, RPM	45890	50410	40680	44290
AIR FLOW				
LFE DIFF PR, IN H2O	.42	.44	.40	.42
LFE PR, IN H2O	.15	.20	.15	.15
LFE TEMP, F	72	73	73	73
PCF	.987	.987	.987	.987
TCF	.9934	.9901	.9901	.9901
AIR RATE, LB/HR	218.41	228.02	207.32	217.68
FUEL FLOW				
TIME FOR 1 LB, SEC	273.2	276.4	271.6	275.2
FUEL RATE, LB/HR	13.18	13.02	13.25	13.08
BSFC,LB/HP.HR	.425	.420	.428	.422
TEMPERATURES				
COOLANT IN, F	70	70	74	76
COOLANT OUT, F	181	181	181	181
OIL SUMP, F	205	205	206	207
AMB AIR, F	72	73	73	73
COMP INLET, F	71	71	71	72
COMP OUTLET, F	142	150	141	144
TURBO INLET, F	924	909	972	989
TURBO OUTLET, F	823	803	865	856
PRESSURES				
COMP INLET, IN H2O	.80	.75	.75	.70
COMP OUTLET, IN HG	4.60	5.80	3.40	4.10
TURBO INLET, IN HG	2.80	3.80	2.00	2.70
TURBO OUTLET, IN HG	.05	.05	.03	.05
AIR-FUEL RATIO	16.57	17.51	15.64	16.64
ENGINE VOL EFF, %	70.1	71.6	68.8	71.1
ENGINE BR TH EFF, %	32.5	32.9	32.3	32.7
EXH-INTAKE PR RATIO	.947	.943	.957	.958
COMP PR BOOST	1.16	1.20	1.12	1.14
VALUE OF YC	.043	.053	.032	.038
COMP TEMP DIFF, F	71	79	70	72
TURBO TEMP DIFF, F	101	106	107	133
COMP ISENTROPIC EFF, %	31.9	35.6	24.3	25.3

STOP

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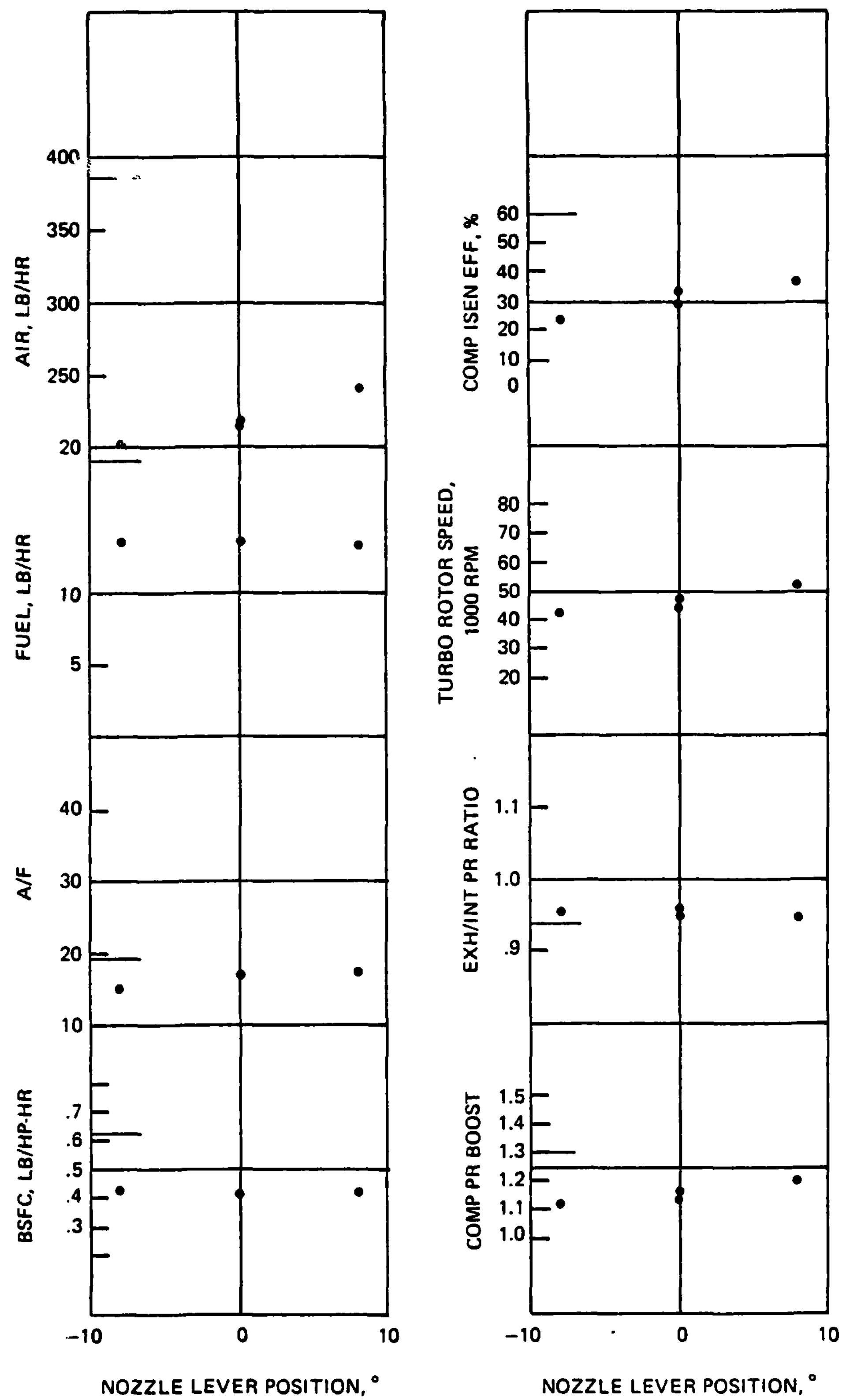


FIGURE D-4 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 1000 RPM AND 124 LB LOAD

TABLE D-5

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger — DEC 11, 1978

BARO PR, IN HG	29.60	29.60	29.60	29.60
DRY BULB TEMP, F	77	77	77	77
WET BULB, F	62	62	62	62
ENGINE SPEED, RPM	1500	1500	1500	1500
DYNO LOAD, LB	26.50	26.50	26.50	26.50
POWER OUTPUT, HP	9.94	9.94	9.94	9.94
BMEP, PSI	21.97	21.97	21.97	21.97
TURBO NOZZLE POS, DEG	0.0	8.0	-8.0	0.0
TURBO ROTOR SPEED, RPM	35960	45290	28930	35050
AIR FLOW				
LFE DIFF PR, IN H2O	.58	.63	.55	.58
LFE PR, IN H2O	.25	.30	.30	.25
LFE TEMP, F	77	79	73	72
PCF	.989	.989	.989	.989
TCF	.9770	.9706	.9901	.9934
AIR RATE, LB/HR	297.18	320.64	285.54	302.15
FUEL FLOW				
TIME FOR 1 LB, SEC	569.6	557.6	588.0	573.2
FUEL RATE, LB/HR	6.32	6.46	6.12	6.28
BSFC,LB/HP.HR	.636	.650	.616	.632
TEMPERATURES				
COOLANT IN, F	80	75	73	82
COOLANT OUT, F	172	176	176	177
OIL SUMP, F	168	192	196	199
AMB AIR, F	77	79	73	72
COMP INLET, F	79	78	71	77
COMP OUTLET, F	111	120	98	108
TURBO INLET, F	464	457	468	464
TURBO OUTLET, F	418	398	426	422
PRESSURES				
COMP INLET, IN H2O	1.50	1.60	1.30	1.50
COMP OUTLET, IN HG	.60	2" 3.10	-1.00	.40
TURBO INLET, IN HG	2.80	5.20 2.5	1.50	2.60
TURBO OUTLET, IN HG	.01	.01	.01	.01
AIR-FUEL RATIO	47.02	49.66	46.64	48.11
ENGINE VOL EFF, %	68.2	69.0	67.6	69.4
ENGINE BR TH EFF, %	21.7	21.3	22.4	21.9
EXH-INTAKE PR RATIO	1.073	1.064	1.087	1.073
COMP PR BOOST	1.02	1.11	.97	1.02
VALUE OF YC	.007	.030	-.009	.005
COMP TEMP DIFF, F	32	42	27	31
TURBO TEMP DIFF, F	46	59	42	42
COMP ISENTROPIC EFF, %	11.5	38.4	-17.4	8.5

STOP

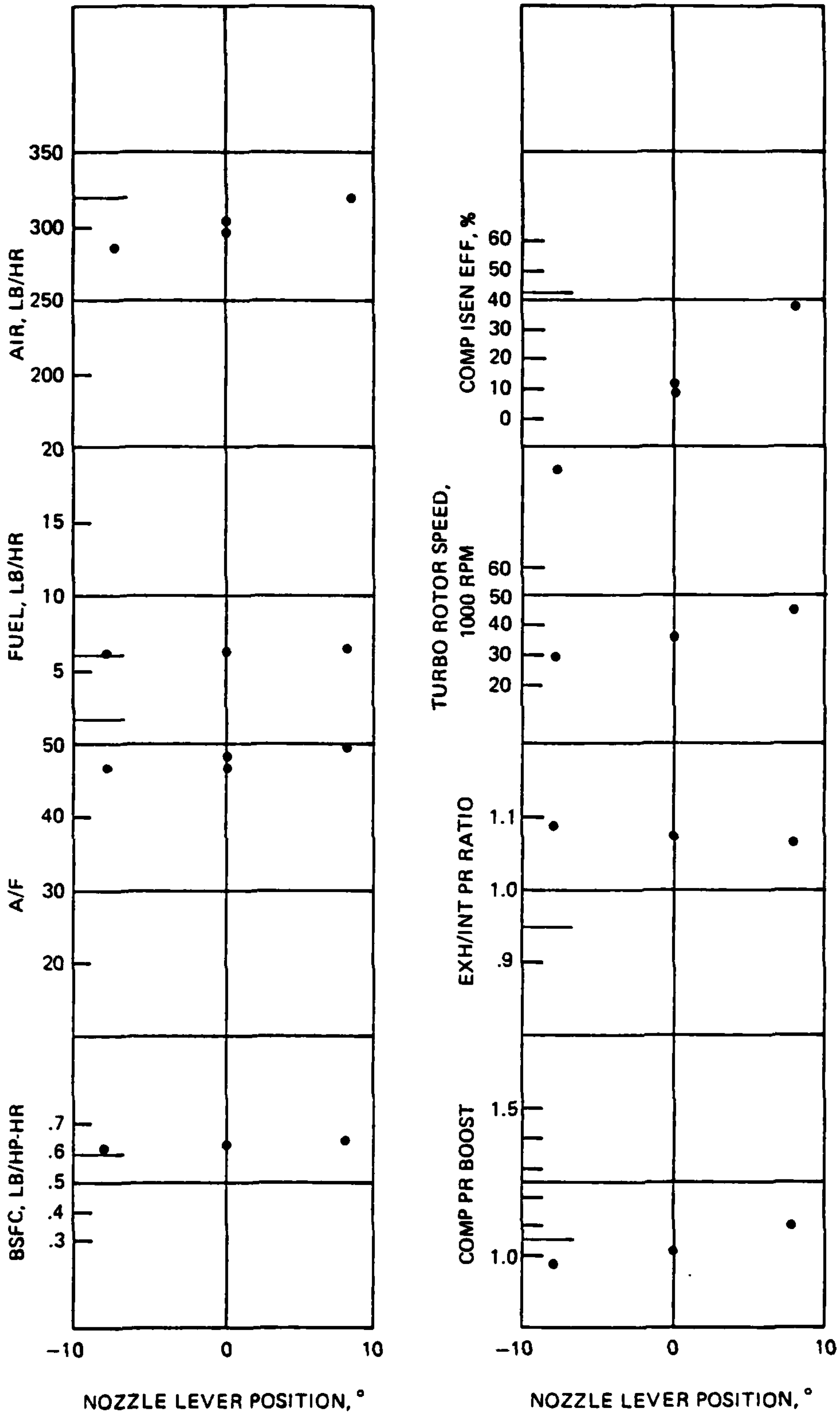


FIGURE D-5 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 1500 RPM AND 26.5 LB LOAD

TABLE D-6

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger

DEC 11 1978

BARO PR, IN HG	29.58	29.58	29.58	29.58
DRY BULB TEMP, F	78	78	78	78
WET BULB, F	62	62	62	62
ENGINE SPEED, RPM	1500	1500	1500	1500
DYNO LOAD, LB	53.00	53.00	53.00	53.00
POWER OUTPUT, HP	19.88	19.88	19.88	19.88
BMEP, PSI	43.94	43.94	43.94	43.94
TURBO NOZZLE POS, DEG	0.0	8.0	-8.0	0.0
TURBO ROTOR SPEED, RPM	40550	49070	34170	39030
AIR FLOW				
LFE DIFF PR, IN H2O	.60	.65	.57	.60
LFE PR, IN H2O	.30	.30	.30	.30
LFE TEMP, F	78	76	73	73
PCF	.988	.988	.988	.988
TCF	.9738	.9802	.9901	.9901
AIR RATE, LB/HR	306.17	333.88	295.72	311.28
FUEL FLOW				
TIME FOR 1 LB, SEC	394.0	382.8	382.8	384.4
FUEL RATE, LB/HR	9.14	9.40	9.40	9.37
BSFC,LB/HP.HR	.460	.473	.473	.471
TEMPERATURES				
COOLANT IN, F	81	79	68	66
COOLANT OUT, F	179	178	178	177
OIL SUMP, F	203	204	204	204
AMB AIR, F	78	76	73	73
COMP INLET, F	79	75	74	74
COMP OUTLET, F	120	129	111	114
TURBO INLET, F	603	583	613	602
TURBO OUTLET, F	524	503	548	530
PRESSURES				
COMP INLET, IN H2O	1.50	1.70	1.40	1.50
COMP OUTLET, IN HG	1.70	4.40	.30	1.50
TURBO INLET, IN HG	3.10	5.30	2.00	2.90
TURBO OUTLET, IN HG	.01	.01	.01	.01
AIR-FUEL RATIO	33.51	35.50	31.44	33.24
ENGINE VOL EFF, %	68.9	70.2	68.6	69.8
ENGINE BR TH EFF, %	30.1	29.2	29.2	29.3
EXH-INTAKE PR RATIO	1.045	1.026	1.057	1.045
COMP PR BOOST	1.06	1.15	1.01	1.05
VALUE OF YC	.017	.042	.004	.015
COMP TEMP DIFF, F	41	54	37	40
TURBO TEMP DIFF, F	79	80	65	72
COMP ISENTROPIC EFF, %	22.5	41.2	5.6	20.4

STOP

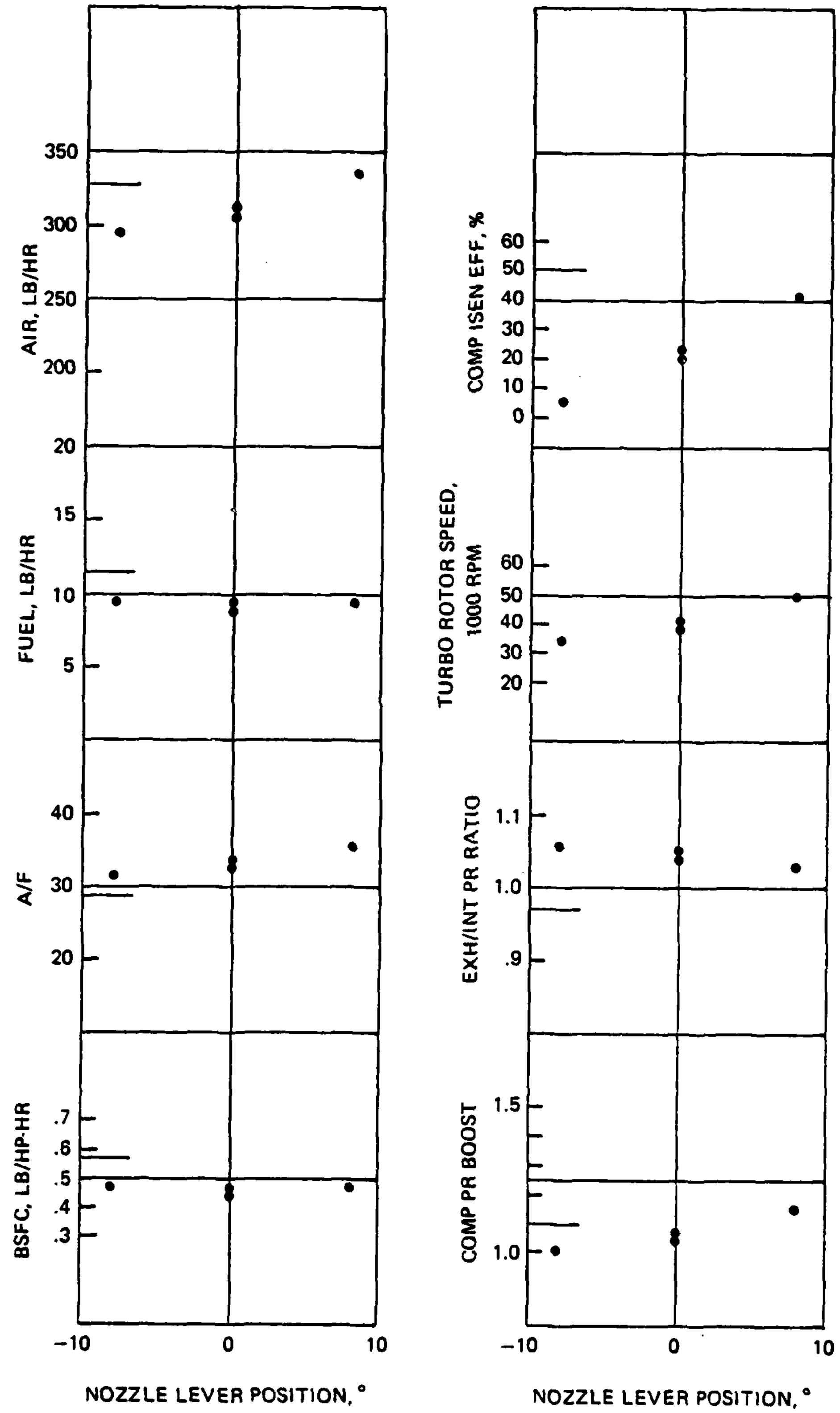


FIGURE D-6 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 1500 RPM AND 53 LB LOAD

TABLE D-7

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger DEC 11 1978

BARO PR, IN HG	29.58	29.58	29.58	29.58
DRY BULB TEMP, F	74	74	74	74
WET BULB, F	59	59	59	59
ENGINE SPEED, RPM	1500	1500	1500	1500
DYNO LOAD, LB	79.00	79.00	79.00	79.00
POWER OUTPUT, HP	29.63	29.63	29.63	29.63
BMEP, PSI	65.50	65.50	65.50	65.50
TURBO NOZZLE POS, DEG	0.0	8.0	-8.0	0.0
TURBO ROTOR SPEED, RPM	45050	53370	40480	43280
AIR FLOW				
LFE DIFF PR, IN H20	.62	.67	.60	.62
LFE PR, IN H20	.30	.35	.30	.30
LFE TEMP, F	74	75	76	76
PCF	.988	.988	.988	.988
TCF	.9868	.9835	.9802	.9802
AIR RATE, LB/HR	320.59	345.25	308.20	318.47
FUEL FLOW				
TIME FOR 1 LB, SEC	294.8	295.6	296.8	293.6
FUEL RATE, LB/HR	12.21	12.18	12.13	12.26
BSFC, LB/HP.HR	.412	.411	.409	.414
TEMPERATURES				
COOLANT IN, F	71	73	75	76
COOLANT OUT, F	179	179	179	179
OIL SUMP, F	208	209	210	211
AMB AIR, F	74	75	76	76
COMP INLET, F	75	76	77	78
COMP OUTLET, F	128	139	125	129
TURBO INLET, F	732	709	758	750
TURBO OUTLET, F	642	612	673	645
PRESSURES				
COMP INLET, IN H20	1.60	1.80	1.50	1.60
COMP OUTLET, IN HG	3.30	5.80	2.00	2.70
TURBO INLET, IN HG	3.50	5.30	2.60	3.10
TURBO OUTLET, IN HG	.05	.05	.05	.05
AIR-FUEL RATIO	26.25	28.35	25.41	25.97
ENGINE VOL EFF, %	69.6	70.9	69.3	70.5
ENGINE BR TH EFF, %	33.5	33.6	33.7	33.4
EXH-INTAKE PR RATIO	1.006	.986	1.019	1.012
COMP PR BOOST	1.12	1.20	1.07	1.10
VALUE OF YC	.032	.054	.020	.026
COMP TEMP DIFF, F	53	63	48	51
TURBO TEMP DIFF, F	90	97	85	105
COMP ISENTROPIC EFF, %	32.1	45.7	22.3	27.8

STOP

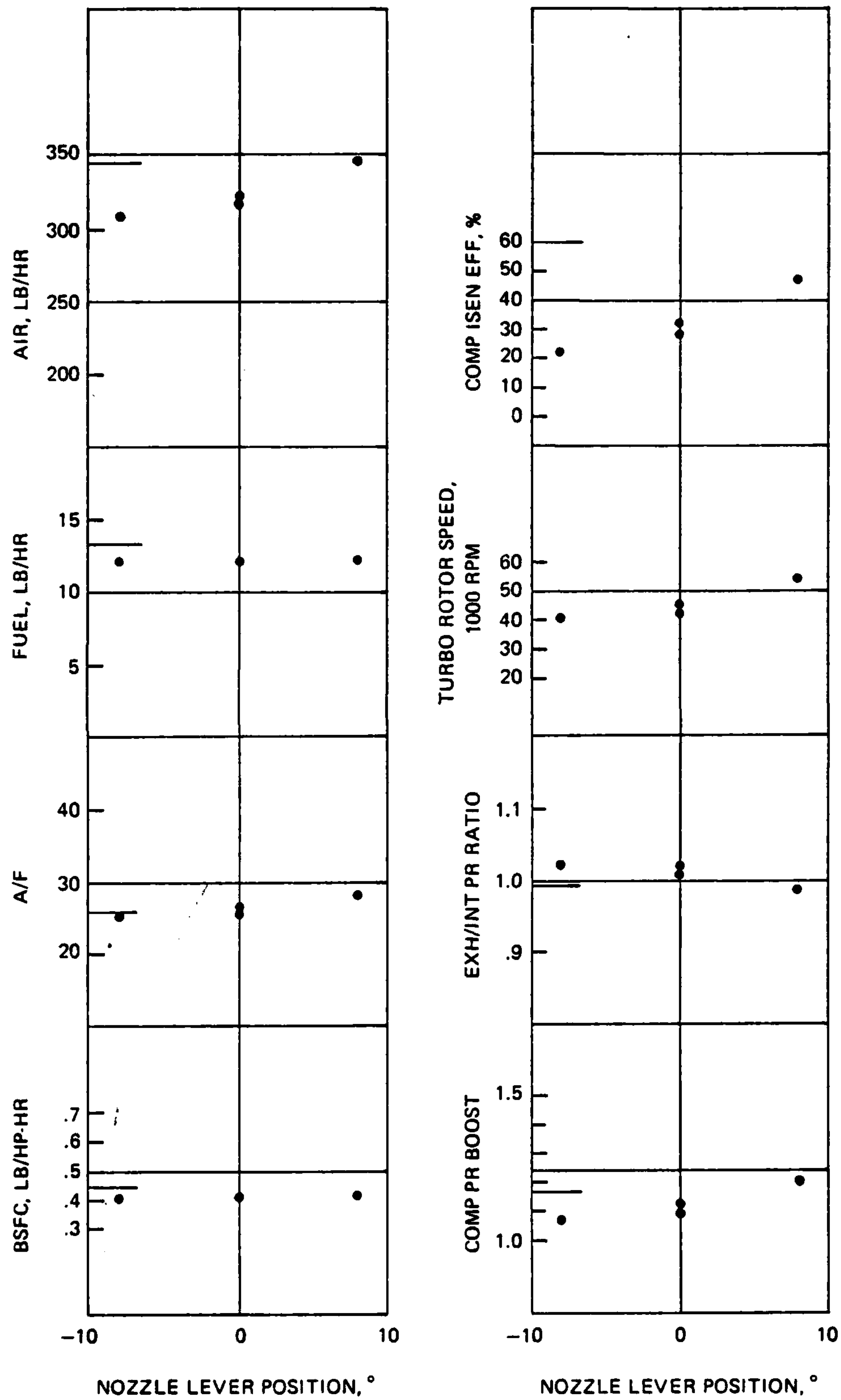


FIGURE D-7 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 1500 RPM AND 79 LB LOAD

TABLE D-8

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger DEC 11, 1978

BARO PR, IN HG	29.59	29.58	29.58	29.58
DRY BULB TEMP, F	77	77	77	77
WET BULB, F	62	62	62	62
ENGINE SPEED, RPM	1500	1500	1500	1500
DYNO LOAD, LB	106.00	106.00	106.00	106.00
POWER OUTPUT, HP	39.75	39.75	39.75	39.75
BMEP, PSI	87.89	87.88	87.88	87.88
TURBO NOZZLE POS, DEG	0.0	8.0	-8.0	0.0
TURBO ROTOR SPEED, RPM	49010	49040	46550	4700
AIR FLOW				
LFE DIFF PR, IN H20	.64	.64	.62	.63
LFE PR, IN H20	.35	.35	.30	.35
LFE TEMP, F	77	77	76	76
PCF	.988	.988	.988	.988
TCF	.9770	.9770	.9802	.9802
AIR RATE, LB/HR	327.62	327.62	318.47	323.57
FUEL FLOW				
TIME FOR 1 LB, SEC	231.6	232.4	227.6	228.4
FUEL RATE, LB/HR	15.54	15.49	15.82	15.76
BSFC,LB/HP.HR	.391	.390	.398	.397
TEMPERATURES				
COOLANT IN, F	80	81	71	69
COOLANT OUT, F	180	180	179	179
OIL SUMP, F	213	216	216	216
AMB AIR, F	77	77	76	76
COMP INLET, F	78	78	77	77
COMP OUTLET, F	141	141	138	138
TURBO INLET, F	865	870	881	879
TURBO OUTLET, F	769	775	792	789
PRESSURES				
COMP INLET, IN H20	1.60	1.65	1.60	1.60
COMP OUTLET, IN HG	4.40	4.40	3.70	3.80
TURBO INLET, IN HG	3.60	3.60	3.20	3.35
TURBO OUTLET, IN HG	.05	.05	.05	.05
AIR-FUEL RATIO	21.08	21.15	20.13	20.53
ENGINE VOL EFF, %	70.3	70.3	69.4	70.3
ENGINE BR TH EFF, %	35.3	35.5	34.7	34.8
EXH-INTAKE PR RATIO	.976	.976	.985	.987
COMP PR BOOST	1.15	1.15	1.13	1.13
VALUE OF YC	.041	.042	.035	.036
COMP TEMP DIFF, F	63	63	61	61
TURBO TEMP DIFF, F	96	95	89	90
COMP ISENTROPIC EFF, %	35.4	35.5	31.1	31.9

STOP

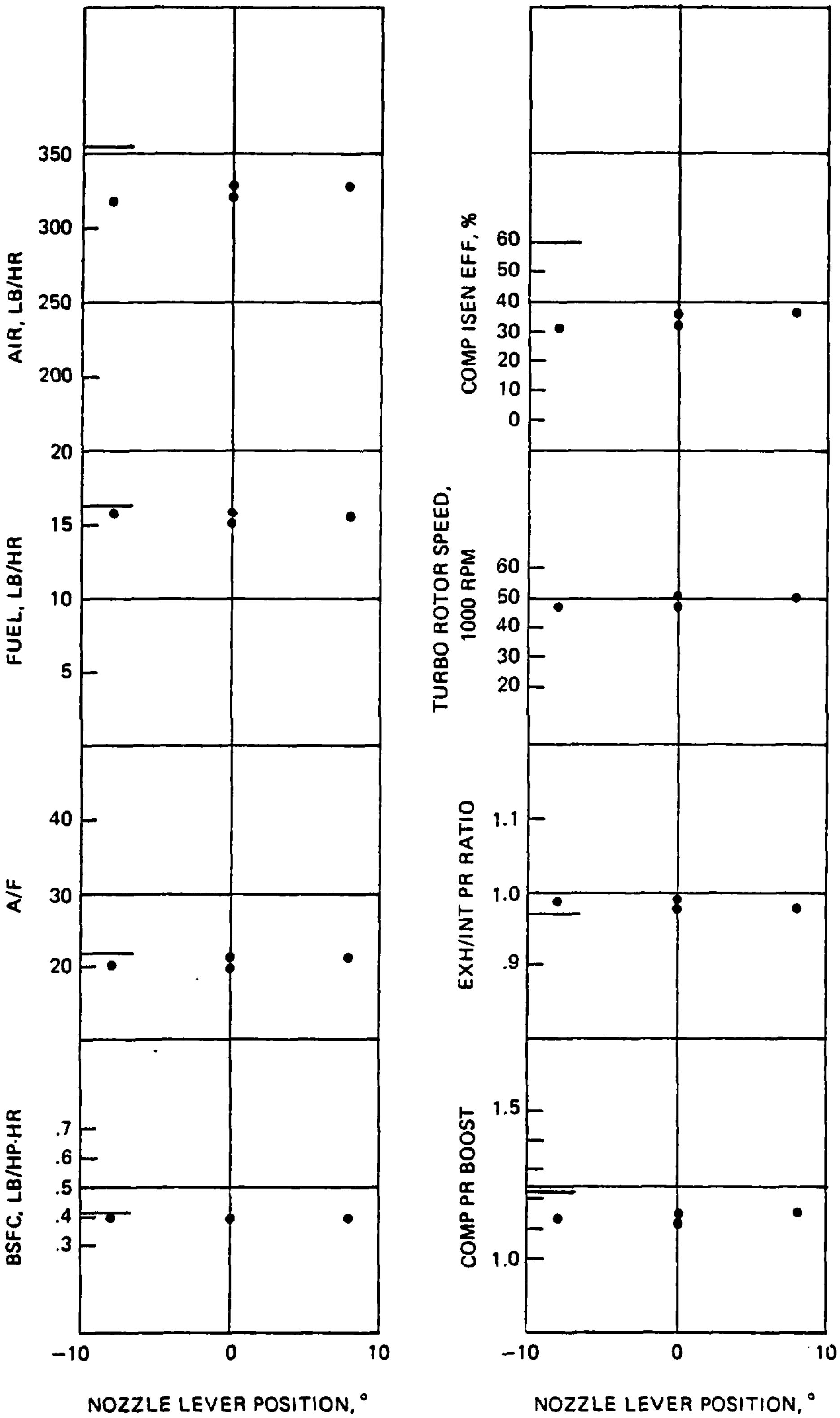


FIGURE D-8 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 1500 RPM AND 106 LB LOAD

TABLE D-9

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger DEC 12,1978

BARO PR, IN HG	29.53	29.53	29.53	29.53
DRY BULB TEMP, F	73	73	73	73
WET BULB, F	58	58	58	58
ENGINE SPEED, RPM	2000	2000	2000	2000
DYNO LOAD, LB	35.00	35.00	35.00	35.00
POWER OUTPUT, HP	17.50	17.50	17.50	17.50
BMEP, PSI	29.02	29.02	29.02	29.02
TURBO NOZZLE POS, DEG	0.0	10.0	-10.0	0.0
TURBO ROTOR SPEED, RPM	49810	70130	37370	50410
AIR FLOW				
LFE DIFF PR, IN H2O	.78	.94	.70	.78
LFE PR, IN H2O	.35	.45	.30	.35
LFE TEMP, F	73	73	73	72
PCF	.986	.986	.986	.986
TCF	.9901	.9901	.9901	.9934
AIR RATE, LB/HR	403.93	486.67	362.55	405.28
FUEL FLOW				
TIME FOR 1 LB, SEC	335.6	326.8	336.8	334.4
FUEL RATE, LB/HR	10.73	11.02	10.69	10.77
BSFC,LB/HP.HR	.613	.629	.611	.615
TEMPERATURES				
COOLANT IN, F	70	70	71	74
COOLANT OUT, F	178	179	178	178
OIL SUMP, F	211	213	213	213
AMB AIR, F	73	73	73	72
COMP INLET, F	73	74	72	72
COMP OUTLET, F	122	158	109	121
TURBO INLET, F	593	563	624	586
TURBO OUTLET, F	534	473	570	508
PRESSURES				
COMP INLET, IN H2O	2.25	2.90	2.00	2.30
COMP OUTLET, IN HG	1.80	9.80	-1.40	1.90
TURBO INLET, IN HG	4.90	12.85	2.50	.50
TURBO OUTLET, IN HG	.01	.10	.10	.01
AIR-FUEL RATIO	37.66	44.18	33.92	37.65
ENGINE VOL EFF, %	68.3	69.6	66.7	68.2
ENGINE BR TH EFF, %	22.5	21.9	22.6	22.5
EXH-INTAKE PR RATIO	1.099	1.078	1.139	.955
COMP PR BOOST	1.07	1.34	.96	1.07
VALUE OF YC	.019	.087	-.012	.020
COMP TEMP DIFF, F	49	84	37	49
TURBO TEMP DIFF, F	59	90	54	78
COMP ISENTROPIC EFF, %	20.3	55.5	-17.7	21.3

STOP

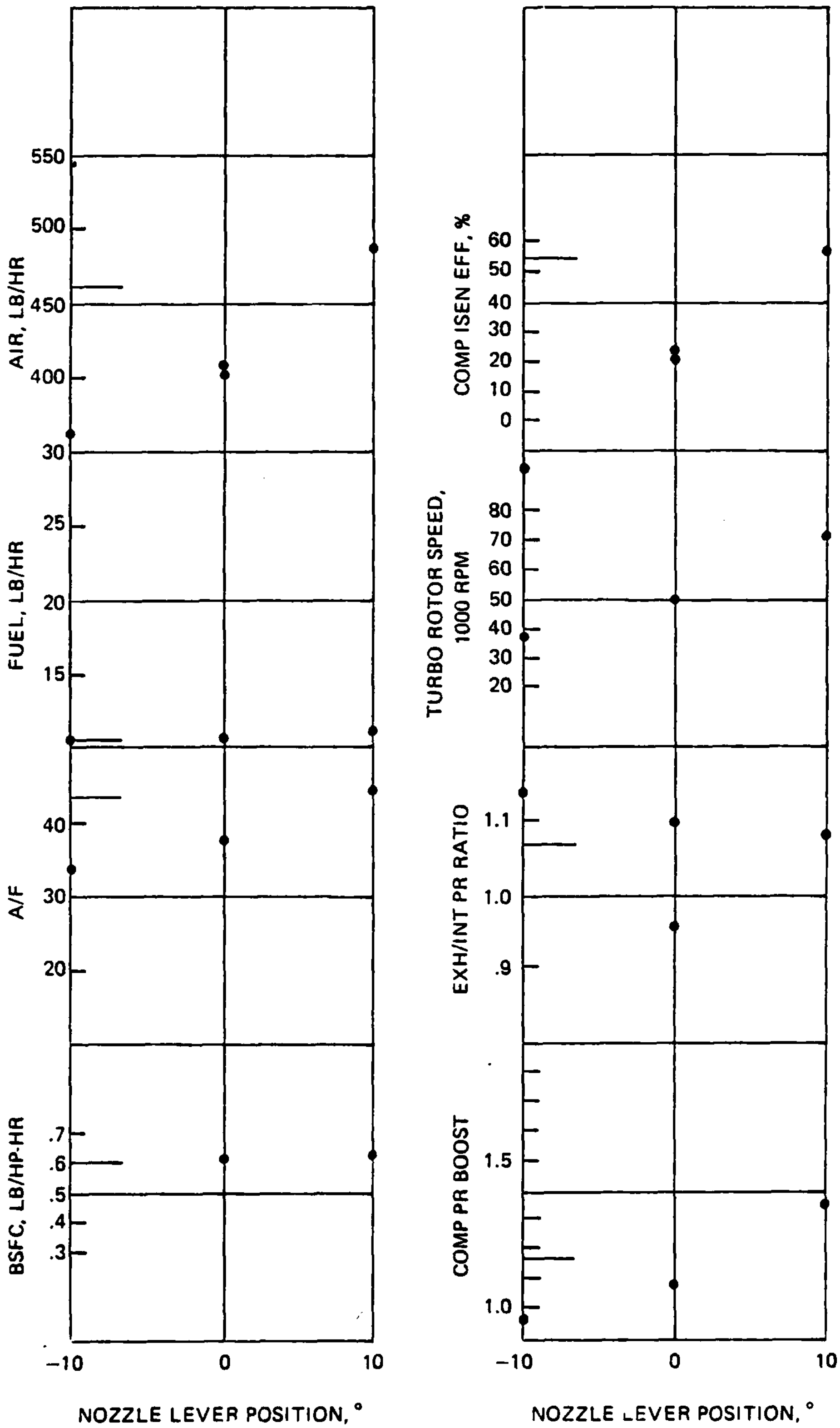


FIGURE D-9 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 2000 RPM AND 35 LB LOAD

TABLE D-10

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger DEC 12, 1978

BARO PR, IN HG	29.53	29.53	29.53	29.53
DRY BULB TEMP, F	71	71	71	71
ET BULB, F	59	59	59	59
NGINE SPEED, RPM	2000	2000	2000	2000
DYNO LOAD, LB	70.50	70.50	70.50	70.50
POWER OUTPUT, HP	35.25	35.25	35.25	35.25
BMEP, PSI	58.45	58.45	58.45	58.45
TURBO NOZZLE POS, DEG	0.0	10.0	-10.0	0.0
TURBO ROTOR SPEED, RPM	60700	79650	47900	56960
AIR FLOW				
LFE DIFF PR, IN H2O	.86	1.04	.76	.83
LFE PR, IN H2O	.40	.50	.35	.35
LFE TEMP, F	71	71	71	71
PCF	.986	.986	.986	.986
TCF	.9967	.9967	.9967	.9967
AIR RATE, LB/HR	448.29	541.98	396.21	432.70
FUEL FLOW				
TIME FOR 1 LB, SEC	226.4	224.0	225.2	227.6
FUEL RATE, LB/HR	15.90	16.07	15.99	15.82
BSFC,LB/HP.HR	.451	.456	.453	.449
TEMPERATURES				
COOLANT IN, F	73	76	73	73
COOLANT OUT, F	179	179	179	179
OIL SUMP, F	215	217	218	218
AMB AIR, F	71	71	71	71
COMP INLET, F	71	72	71	71
COMP OUTLET, F	141	182	125	136
TURBO INLET, F	736	691	792	759
TURBO OUTLET, F	655	571	720	684
PRESSURES				
COMP INLET, IN H2O	2.60	3.40	2.15	2.40
COMP OUTLET, IN HG	5.80	15.00	1.40	4.40
TURBO INLET, IN HG	6.70	15.40	3.70	5.65
TURBO OUTLET, IN HG	.10	.15	.10	.10
AIR-FUEL RATIO	28.19	33.72	24.79	27.36
ENGINE VOL EFF, %	69.4	71.1	68.2	69.2
ENGINE BR TH EFF, %	30.6	30.3	30.5	30.8
EXH-INTAKE PR RATIO	1.025	1.009	1.074	1.037
COMP PR BOOST	1.20	1.52	1.05	1.16
VALUE OF YC	.054	.127	.015	.042
COMP TEMP DIFF, F	70	110	54	65
TURBO TEMP DIFF, F	81	120	72	75
COMP ISENTROPIC EFF, %	41.3	61.4	14.6	34.4

STOP

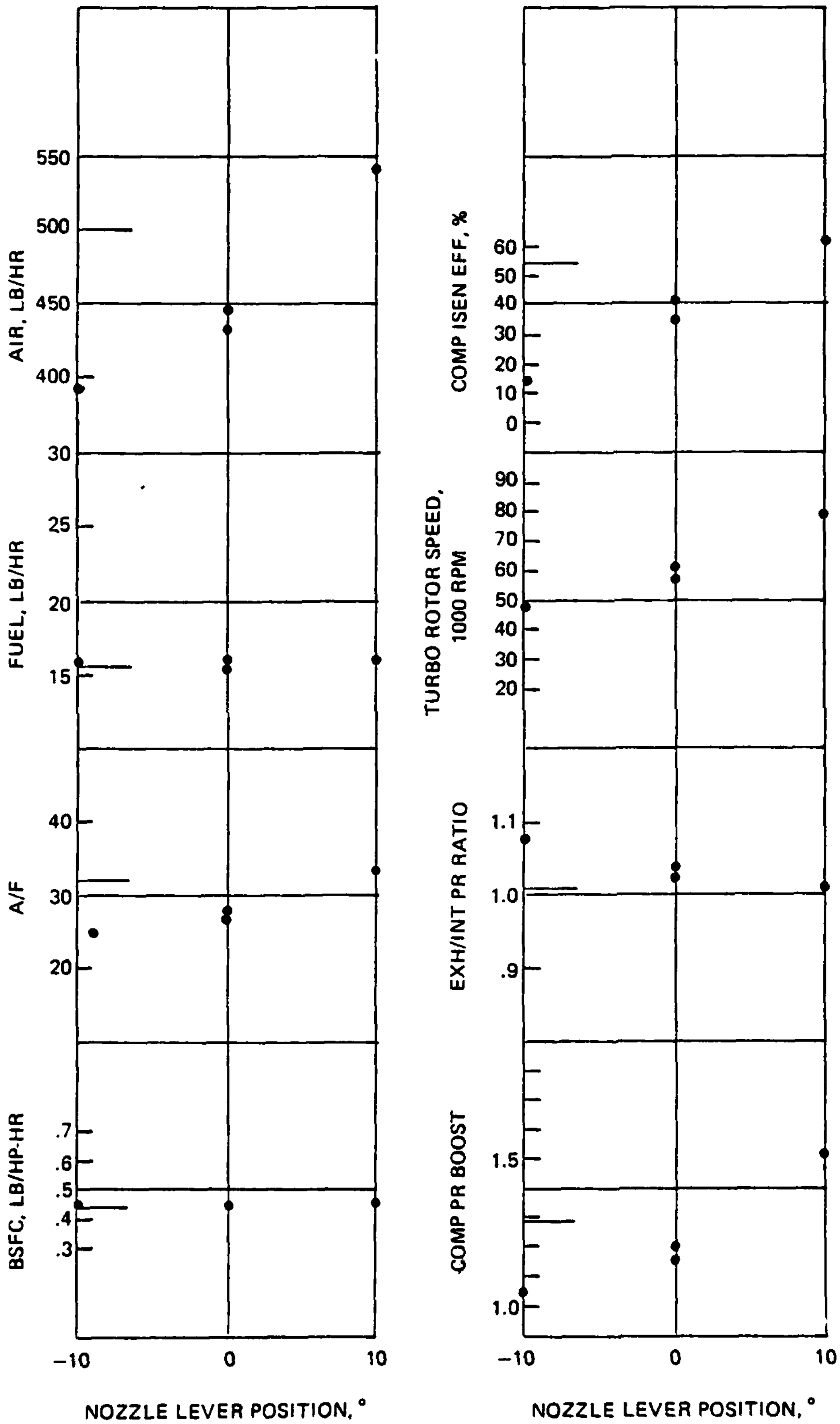


FIGURE D-10 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 2000 RPM AND 70.5 LB LOAD

TABLE D-11

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger

DEC 12, 1978

BARO PR, IN HG	29.53	29.55	29.53	29.53
DRY BULB TEMP, F	71	71	71	71
WET BULB, F	59	59	59	59
ENGINE SPEED, RPM	2000	2000	2000	2000
DYNO LOAD, LB	105.00	105.00	105.00	105.00
POWER OUTPUT, HP	52.50	52.50	52.50	52.50
BMEP, PSI	87.06	87.06	87.06	87.06
TURBO NOZZLE POS, DEG	0.0	10.0	10.0	-10.0
TURBO ROTOR SPEED, RPM	65910	67210	88780	55220
AIR FLOW				
LFE DIFF PR, IN H20	.90	.91	1.16	.80
LFE PR, IN H20	.40	.40	.50	.35
LFE TEMP, F	71	73	73	70
PCF	.986	.986	.986	.986
TCF	.9967	.9901	.9901	1.0000
AIR RATE, LB/HR	469.14	471.20	600.50	418.46
	0		10	-10
FUEL FLOW				
TIME FOR 1 LB, SEC	169.6	169.6	172.4	169.2
FUEL RATE, LB/HR	21.23	21.23	20.88	21.28
BSFC, LB/HP.HR	.404	.404	.398	.405
TEMPERATURES				
COOLANT IN, F	70	71	69	71
COOLANT OUT, F	180	180	180	180
OIL SUMP, F	221	225	223	225
AMB AIR, F	71	73	73	70
COMP INLET, F	72	73	72	69
COMP OUTLET, F	157	162	216	138
TURBO INLET, F	885	887	785	938
TURBO OUTLET, F	797	799	638	861
PRESSURES				
COMP INLET, IN H20	2.70	2.75	3.90	2.30
COMP OUTLET, IN HG	8.10	8.60	21.80	4.10
TURBO INLET, IN HG	7.20	7.50	19.70	4.65
TURBO OUTLET, IN HG	.15	.15	.15	.15
AIR-FUEL RATIO	22.10	22.20	28.76	19.67
ENGINE VOL EFF, %	70.0	69.9	72.0	67.7
ENGINE BR TH EFF, %	34.2	34.2	34.7	34.1
EXH-INTAKE PR RATIO	.976	.971	.959	1.016
COMP PR BOOST	1.28	1.30	1.76	1.15
VALUE OF YC	.074	.078	.174	.039
COMP TEMP DIFF, F	85	89	144	69
TURBO TEMP DIFF, F	88	88	147	77
COMP ISENTROPIC EFF, %	46.1	46.5	64.3	30.2

STOP

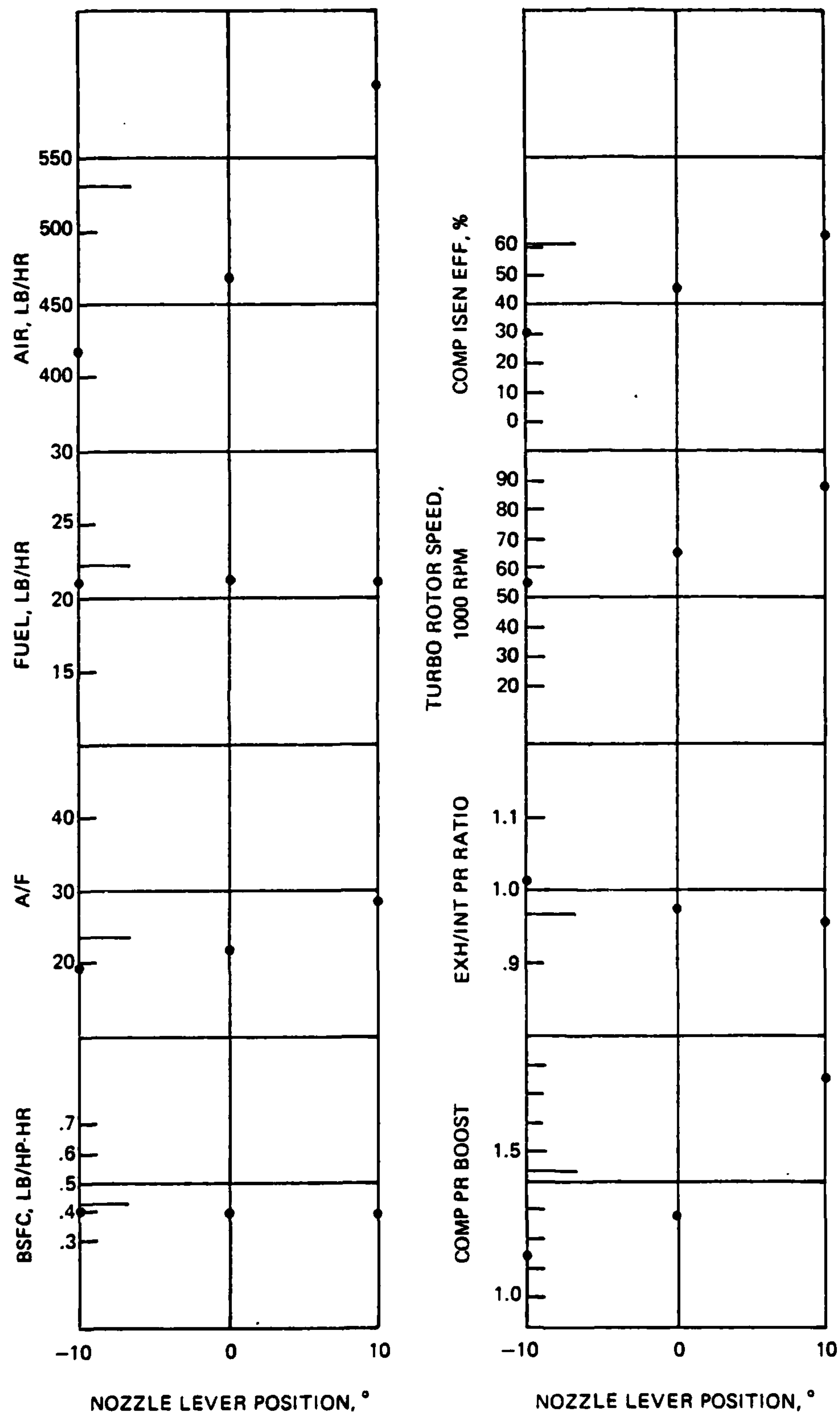


FIGURE D-11 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 2000 RPM AND 105 LB LOAD

TABLE D-12

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger

DEC 12, 1978

BARO PR, IN HG	29.53	29.53	29.53	29.53
DRY BULB TEMP, F	81	81	81	81
WET BULB, F	67	67	67	67
ENGINE SPEED, RPM	2000	2000	2000	2000
DYNO LOAD, LB	140.00	140.00	140.00	140.00
POWER OUTPUT, HP	70.00	70.00	70.00	70.00
BMEP, PSI	116.07	116.07	116.07	116.07
TURBO NOZZLE POS, DEG	0.0	10.0	-10.0	0.0
TURBO ROTOR SPEED, RPM	79440	97040	64700	79570
AIR FLOW				
LFE DIFF PR, IN H20	1.04	1.28	.90	1.04
LFE PR, IN H20	.50	.55	.40	.50
LFE TEMP, F	81	82	82	80
PCF	.986	.986	.986	.986
TCF	.9642	.9611	.9611	.9674
AIR RATE, LB/HR	524.34	643.15	452.39	526.06
FUEL FLOW				
TIME FOR 1 LB, SEC	133.2	138.4	130.8	137.2
FUEL RATE, LB/HR	27.03	26.01	27.52	26.24
BSFC,LB/HP.HR	.386	.372	.393	.375
TEMPERATURES				
COOLANT IN, F	74	80	83	85
COOLANT OUT, F	181	181	181	181
OIL SUMP, F	229	229	227	227
AMB AIR, F	81	82	82	80
COMP INLET, F	82	83	83	80
COMP OUTLET, F	200	259	170	197
TURBO INLET, F	1006	904	1117	990
TURBO OUTLET, F	870	723	964	855
PRESSURES				
COMP INLET, IN H20	3.30	4.50	2.60	3.30
COMP OUTLET, IN HG	14.60	28.10	7.60	14.60
TURBO INLET, IN HG	10.30	23.20	5.70	10.30
TURBO OUTLET, IN HG	.20	.20	.20	.20
AIR-FUEL RATIO	19.40	24.73	16.44	20.05
ENGINE VOL EFF, %	71.4	73.0	69.8	71.3
ENGINE BR TH EFF, %	35.8	37.2	35.1	36.9
EXH-INTAKE PR RATIO	.903	.915	.949	.903
COMP PR BOOST	1.51	1.97	1.27	1.51
VALUE OF YC	.124	.214	.069	.124
COMP TEMP DIFF, F	118	176	87	117
TURBO TEMP DIFF, F	136	181	153	135
COMP ISENTROPIC EFF, %	56.9	86.0	43.3	57.2

STOP

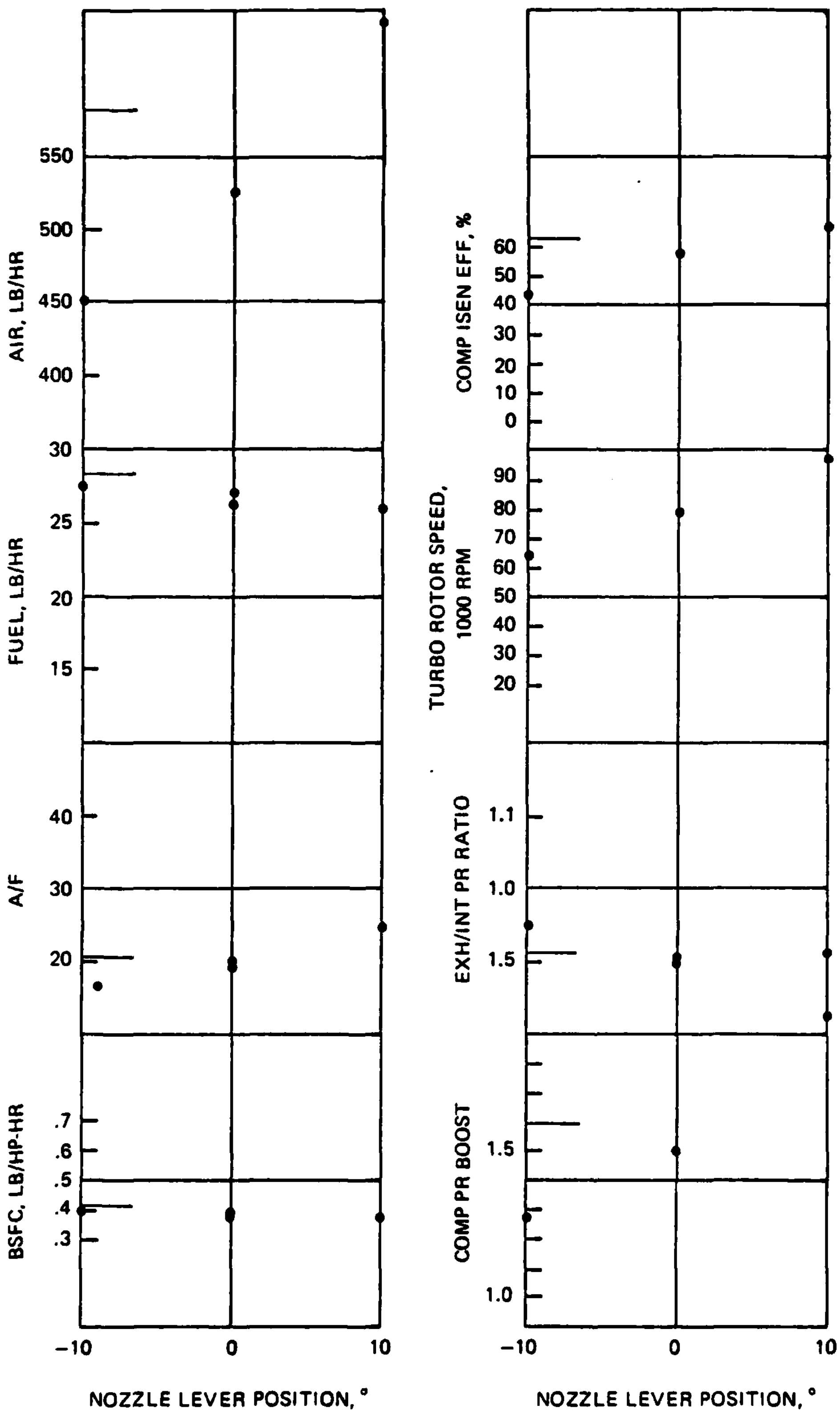


FIGURE D-12 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 2000 RPM AND 140 LB LOAD

TABLE D-13

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger DEC 13 1978

BARO PR, IN HG	29.51	29.51	29.51	29.51
DRY BULB TEMP, F	77	77	77	77
WET BULB, F	61	61	61	61
ENGINE SPEED, RPM	2500	2500	2500	2500
DYNO LOAD, LB	37.00	37.00	37.00	37.00
POWER OUTPUT, HP	23.13	23.13	23.13	23.13
BMEP, PSI	30.68	30.68	30.68	30.68
TURBO NOZZLE POS, DEG	0.0	10.0	-10.0	0.0
TURBO ROTOR SPEED, RPM	64400	87800	49400	65000
AIR FLOW				
LFE DIFF PR, IN H2O	.99	1.28	.86	.99
LFE PR, IN H2O	.50	.60	.40	.50
LFE TEMP, F	73	73	73	73
PCF	.985	.985	.985	.985
TCF	.9901	.9901	.9901	.9901
AIR RATE, LB/HR	512.14	662.00	445.00	512.14
FUEL FLOW				
TIME FOR 1 LB, SEC	234.4	223.4	238.0	238.4
FUEL RATE, LB/HR	15.36	16.11	15.13	15.10
BSFC,LB/HP.HR	.664	.697	.654	.653
TEMPERATURES				
COOLANT IN, F	72	71	71	71
COOLANT OUT, F	178	179	179	179
OIL SUMP, F	219	224	224	224
AMB AIR, F	72	73	73	73
COMP INLET, F	73	73	73	73
COMP OUTLET, F	148	207	125	147
TURBO INLET, F	718	680	776	715
TURBO OUTLET, F	636	542	707	634
PRESSURES				
COMP INLET, IN H2O	3.05	4.60	2.50	3.00
COMP OUTLET, IN HG	4.10	16.80	-1.50	3.90
TURBO INLET, IN HG	8.10	21.70	4.05	8.10
TURBO OUTLET, IN HG	.10	.15	.10	.10
AIR-FUEL RATIO	33.35	41.08	29.42	33.92
ENGINE VOL EFF, %	67.4	69.4	67.7	67.7
ENGINE BR TH EFF, %	20.8	19.8	21.1	21.2
EXH-INTAKE PR RATIO	1.119	1.106	1.198	1.126
COMP PR BOOST	1.15	1.59	.96	1.14
VALUE OF YC	.040	.141	-.013	.038
COMP TEMP DIFF, F	75	134	52	74
TURBO TEMP DIFF, F	82	138	69	81
COMP ISENTROPIC EFF, %	28.4	56.0	-13.3	27.5

STOP

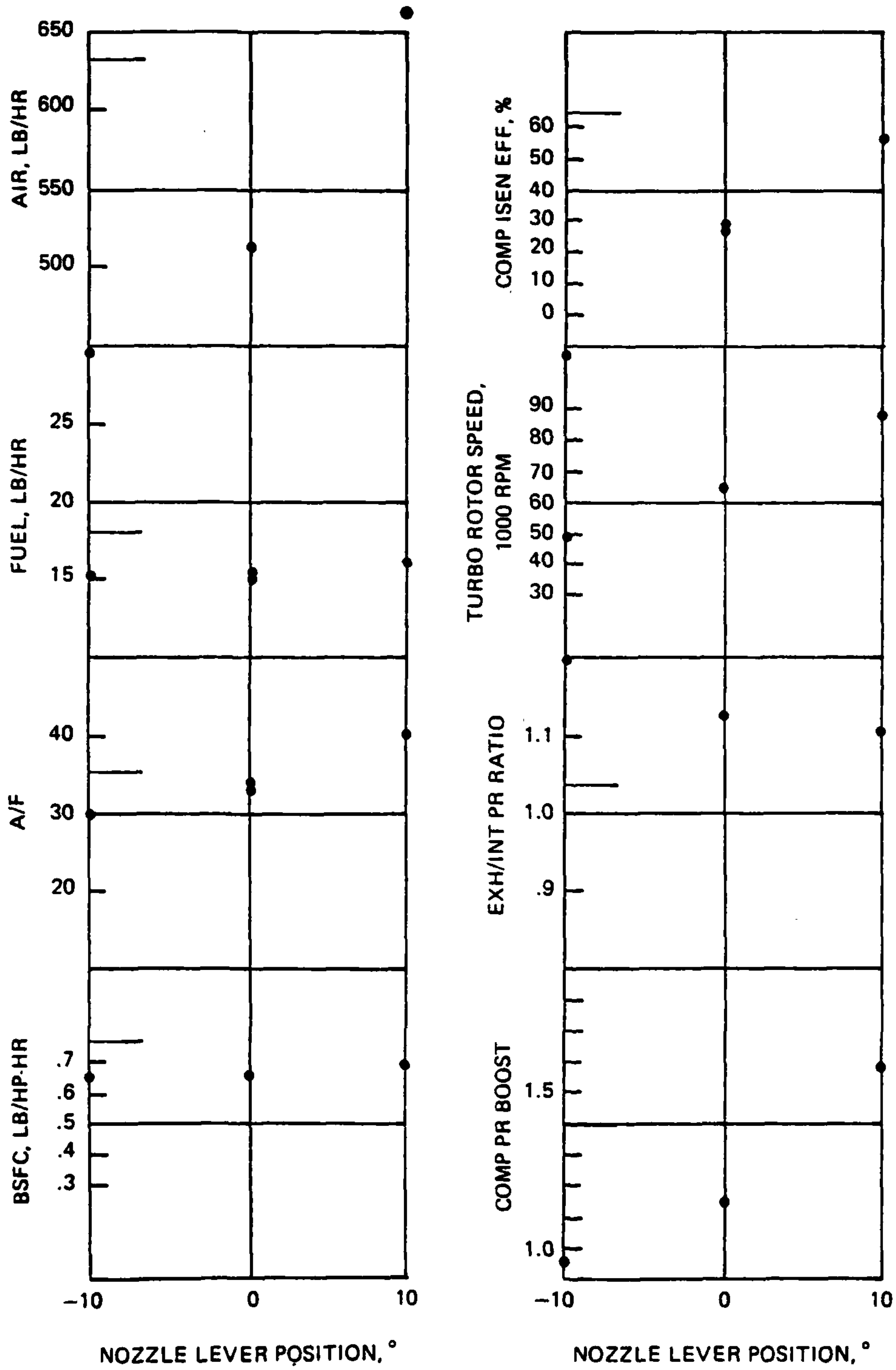


FIGURE D-13 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 2500 RPM AND 37 LB LOAD

TABLE D-14

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger

DEC 13, 1978

BARO PR, IN HG	29.54	29.54	29.54	29.54
DRY BULB TEMP, F	77	77	77	77
WET BULB, F	61	61	61	61
ENGINE SPEED, RPM	2500	2500	2500	2500
DYNO LOAD, LB	73.50	73.50	73.50	73.50
POWER OUTPUT, HP	45.94	45.94	45.94	45.94
BMEP, PSI	60.94	60.94	60.94	60.94
TURBO NOZZLE POS, DEG	0.0	10.0	-10.0	0.0
TURBO ROTOR SPEED, RPM	80000	91500	60900	80000
AIR FLOW				
LFE DIFF PR, IN H2O	1.16	1.46	.96	1.17
LFE PR, IN H2O	.55	.65	.45	.50
LFE TEMP, F	74	74	73	74
PCF	.986	.986	.986	.986
TCF	.9868	.9868	.9901	.9868
AIR RATE, LB/HR	598.63	753.26	497.19	603.87
FUEL FLOW				
TIME FOR 1 LB, SEC	160.0	157.2	160.8	159.6
FUEL RATE, LB/HR	22.50	22.90	22.39	22.56
BSFC,LB/HP.HR	.490	.499	.487	.491
TEMPERATURES				
COOLANT IN, F	71	75	72	75
COOLANT OUT, F	179	180	180	180
OIL SUMP, F	229	231	230	230
AMB AIR, F	74	74	73	74
COMP INLET, F	75	74	74	75
COMP OUTLET, F	185	251	145	187
TURBO INLET, F	876	823	970	880
TURBO OUTLET, F	770	651	891	774
PRESSURES				
COMP INLET, IN H2O	3.80	5.50	2.90	3.90
COMP OUTLET, IN HG	10.70	25.00	2.80	11.10
TURBO INLET, IN HG	11.50	26.70	5.80	11.80
TURBO OUTLET, IN HG	.20	.25	.17	.20
AIR-FUEL RATIO	26.61	32.89	22.21	26.77
ENGINE VOL EFF, %	69.9	71.5	67.7	70.0
ENGINE BR TH EFF, %	28.2	27.7	28.3	28.1
EXH-INTAKE PR RATIO	1.020	1.031	1.093	1.017
COMP PR BOOST	1.38	1.87	1.10	1.39
VALUE OF YC	.095	.196	.028	.098
COMP TEMP DIFF, F	110	177	71	112
TURBO TEMP DIFF, F	106	172	79	106
COMP ISENTROPIC EFF, %	46.2	59.0	21.3	46.9

STOP

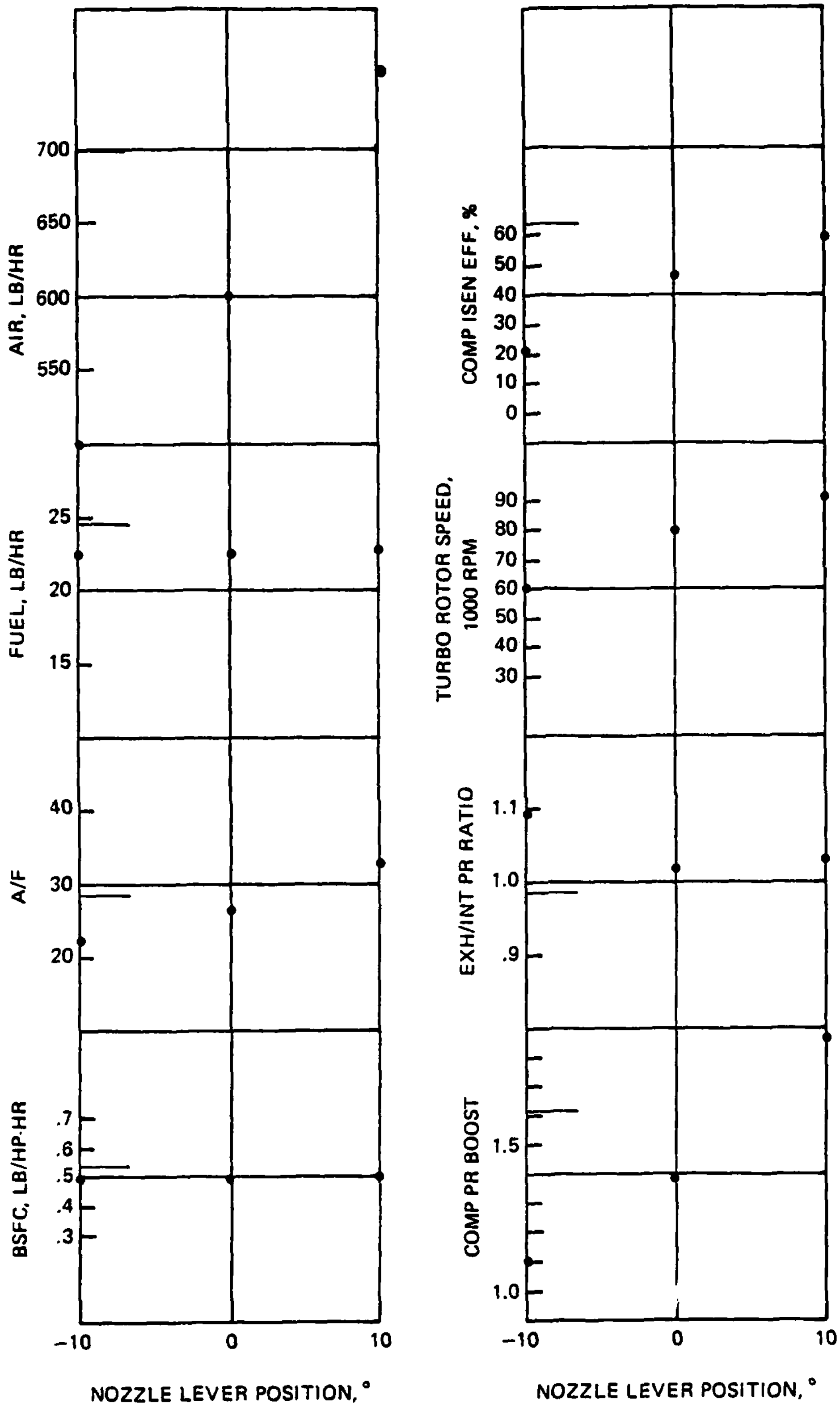


FIGURE D-14 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 2500 RPM AND 73.5 LB LOAD

TABLE D-15

TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger

DEC 13, 1978

BARO PR, IN HG	29.57	29.57	29.57	29.57
DRY BULB TEMP, F	77	77	77	77
WET BULB, F	61	61	61	61
ENGINE SPEED, RPM	2500	2500	2500	2500
DYNO LOAD, LB	111.00	111.00	111.00	111.00
POWER OUTPUT, HP	69.38	69.38	69.38	69.38
BMEP, PSI	92.03	92.03	92.03	92.03
TURBO NOZZLE POS, DEG	0.0	10.0	-10.0	0.0
TURBO ROTOR SPEED, RPM	85900	73200	71500	83800
AIR FLOW				
LFE DIFF PR, IN H2O	1.30	1.53	1.06	1.22
LFE PR, IN H2O	.60	.65	.50	.55
LFE TEMP, F	75	74	74	75
PCF	.987	.987	.987	.987
TCF	.9835	.9868	.9868	.9835
AIR RATE, LB/HR	669.25	790.18	547.65	628.15
FUEL FLOW				
TIME FOR 1 LB, SEC	127.7	127.1	126.4	128.4
FUEL RATE, LB/HR	28.19	28.32	28.48	28.04
BSFC,LB/HP.HR	.406	.408	.411	.404
TEMPERATURES				
COOLANT IN, F	75	77	83	82
COOLANT OUT, F	181	181	181	181
OIL SUMP, F	236	238	236	235
AMB AIR, F	75	74	74	75
COMP INLET, F	75	74	75	76
COMP OUTLET, F	219	281	176	206
TURBO INLET, F	990	930	1010	1025
TURBO OUTLET, F	860	734	1012	911
PRESSURES				
COMP INLET, IN H2O	4.60	6.00	3.40	4.10
COMP OUTLET, IN HG	17.30	29.70	7.10	13.80
TURBO INLET, IN HG	15.20	29.30	7.70	12.30
TURBO OUTLET, IN HG	.30	.35	.25	.25
AIR-FUEL RATIO	23.74	27.90	19.23	22.40
ENGINE VOL EFF, %	70.6	71.9	69.1	70.2
ENGINE BR TH EFF, %	34.0	33.8	33.7	34.2
EXH-INTAKE PR RATIO	.955	.993	1.016	.965
COMP PR BOOST	1.60	2.03	1.25	1.48
VALUE OF YC	.144	.224	.066	.119
COMP TEMP DIFF, F	144	207	101	130
TURBO TEMP DIFF, F	130	196	-2	114
COMP ISENTROPIC EFF, %	53.5	57.9	34.9	48.9

STOP

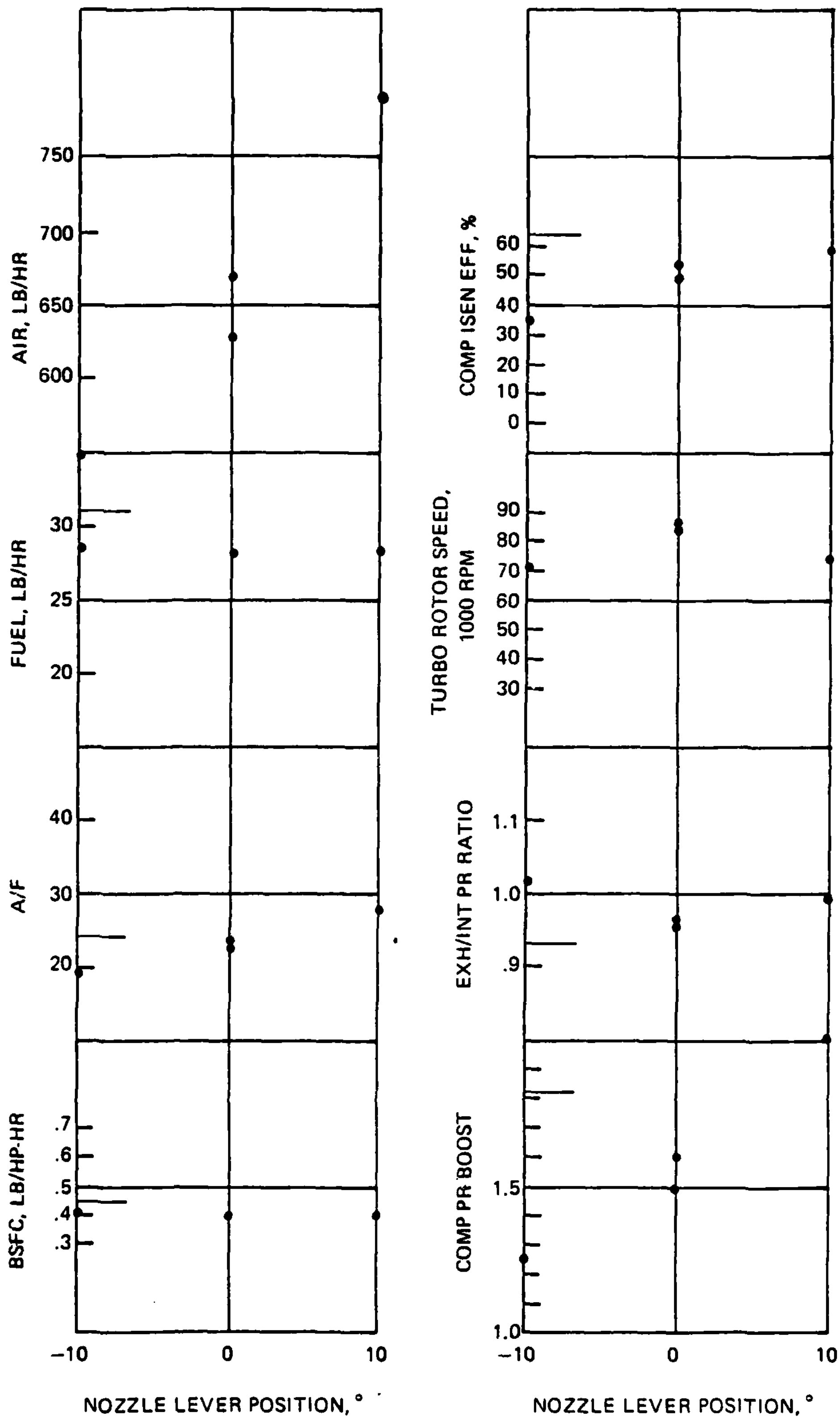


FIGURE D-15 - INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 2500 RPM AND 111 LB LOAD

TABLE D-16

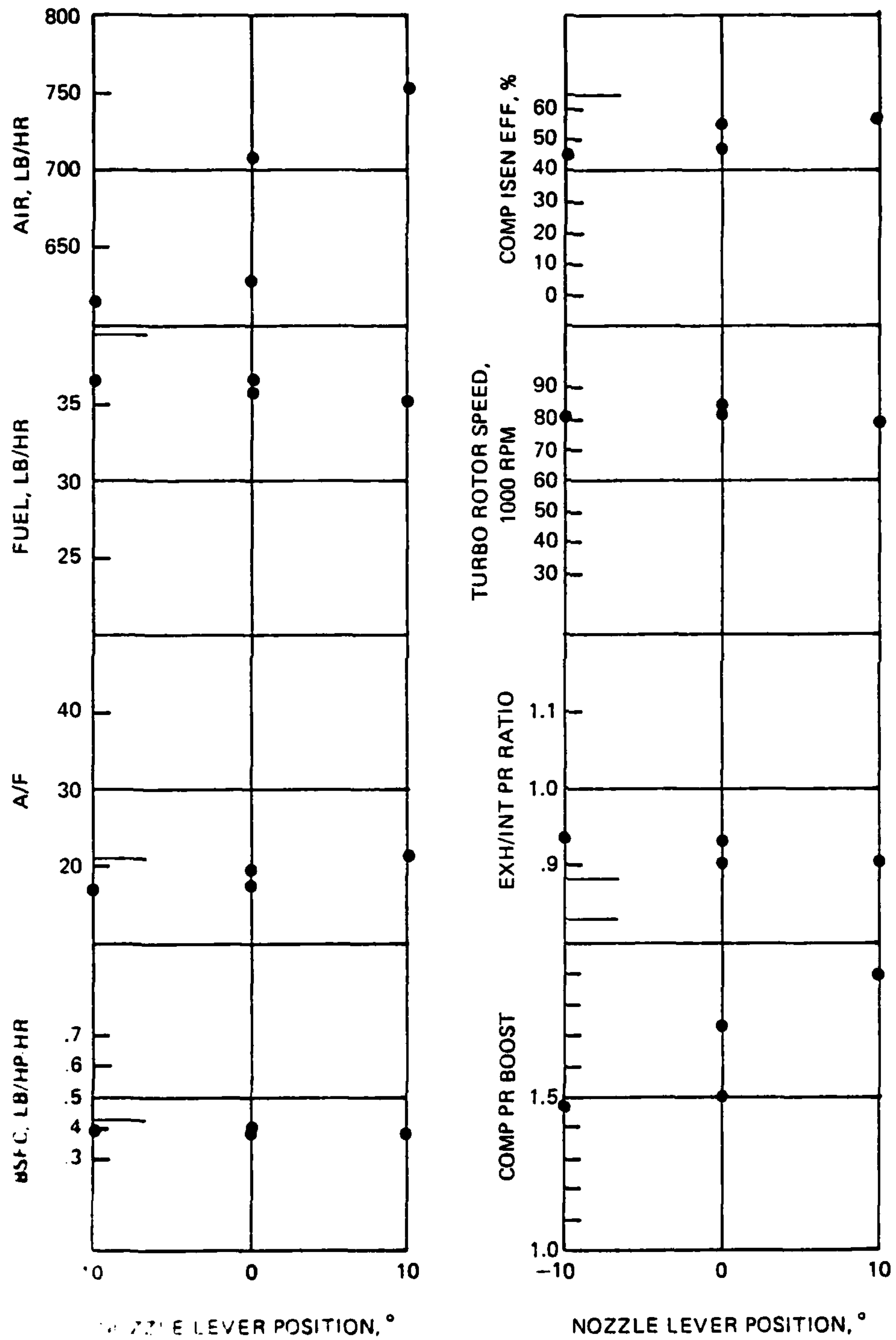
TEST DATA AND RESULTS

ENGINE: DEERE 4239T

Aerodyne Turbocharger DEC 13, 1978

BARO PR, IN HG	29.57	29.57	29.57	29.57
DRY BULB TEMP, F	77	77	77	77
WET BULB, F	61	61	61	61
ENGINE SPEED, RPM	2500	2500	2500	2500
DYNO LOAD, LB	148.00	148.00	148.00	148.00
POWER OUTPUT, HP	92.50	92.50	92.50	92.50
BMEP, PSI	122.71	122.71	122.71	122.71
TURBO NOZZLE POS, DEG	0.0	10.0	-10.0	0.0
TURBO ROTOR SPEED, RPM	85000	78200	81800	82000
AIR FLOW				
LFE DIFF PR, IN H20	1.38	1.47	1.20	1.22
LFE PR, IN H20	.65	.70	.60	.60
LFE TEMP, F	76	76	75	75
PCF	.987	.987	.987	.987
TCF	.9802	.9802	.9835	.9835
AIR RATE, LB/HR	708.00	754.08	617.77	628.07
FUEL FLOW				
TIME FOR 1 LB, SEC	100.5	102.0	98.8	98.6
FUEL RATE, LB/HR	35.82	35.29	36.44	36.51
BSFC,LB/HP.HR	.387	.382	.394	.395
TEMPERATURES				
COOLANT IN, F	91	91	85	82
COOLANT OUT, F	182	182	182	182
OIL SUMP, F	242	241	239	237
AMB AIR, F	76	76	75	75
COMP INLET, F	77	76	76	76
COMP OUTLET, F	242	265	210	214
TURBO INLET, F	1140	1095	1225	1220
TURBO OUTLET, F	967	933	1114	1106
PRESSURES				
COMP INLET, IN H20	5.00	5.60	4.10	4.20
COMP OUTLET, IN HG	21.00	25.60	13.40	14.20
TURBO INLET, IN HG	16.00	20.40	10.70	11.20
TURBO OUTLET, IN HG	.35	.40	.30	.30
AIR-FUEL RATIO	19.77	21.37	16.95	17.20
ENGINE VOL EFF, %	71.5	72.1	70.1	70.4
ENGINE BR TH EFF, %	35.7	36.2	35.1	35.0
EXH-INTAKE PR RATIO	.901	.906	.937	.931
COMP PR BOOST	1.73	1.89	1.47	1.50
VALUE OF YC	.169	.199	.116	.122
COMP TEMP DIFF, F	165	189	134	138
TURBO TEMP DIFF, F	173	162	111	114
COMP ISENTROPIC EFF, %	55.1	56.5	46.3	47.2

STOP



- INFLUENCE OF TURBOCHARGER NOZZLE POSITION ON ENGINE PERFORMANCE AT 2500 RPM AND 148 LB LOAD

APPENDIX E

EMISSIONS TEST DATA AND RESULTS

TABLE E-1

Speed-Load Schedule of 13-Mode Federal Diesel Emission Cycle

Mode	Speed	Torque
1	IDLE	--
2	S	0.02 x Tm
3	S	0.25 x Tm
4	S	0.50 x Tm
5	S	0.75 x Tm
6	S	Tm
7	IDLE	--
8	Sm	T
9	Sm	0.75 x T
10	Sm	0.50 x T
11	Sm	0.25 x T
12	Sm	0.02 x T
13	IDLE	--

NOTES:

Tm - Rated Torque

Sm - Rated Speed

T - Highest Torque at Rated Speed

S - Highest Speed at Rated Torque

TABLE E-2

13-MODE FEDERAL DIESEL EMISSION CYCLE

JOHN DEERE 4239T ENGINE NATURALLY ASPIRATED 25 JULY 1978
PROJECT 11-5214-001 13-MODE NATURAL ASPIRATION TEST

MODE	ENGINE SPEED RPM	TORQUE LB-FT	POWER BHP	FUEL FLOW LB/MIN	AIR FLOW LB/MIN	EXHAUST FLOW LB/MIN	FUEL AIR RATIO
1	839	0.0	0.0	.03	2.70	2.73	.013
2	1700	2.6	.8	.08	5.88	5.96	.013
3	1700	33.5	10.8	.12	5.86	5.98	.021
4	1700	67.0	21.7	.17	5.84	6.01	.030
5	1700	100.4	32.5	.23	5.74	5.97	.041
6	1700	133.9	43.3	.30	5.67	5.97	.052
7	848	0.0	0.0	.04	2.75	2.79	.013
8	2500	98.5	46.9	.38	7.86	8.24	.048
9	2500	73.5	35.0	.31	7.91	8.22	.040
10	2500	49.2	23.4	.26	7.89	8.15	.034
11	2500	24.7	11.7	.23	7.97	8.20	.029
12	2500	2.6	1.3	.22	7.97	8.19	.027
13	822	0.0	0.0	.03	2.71	2.74	.013

MODE	HC PPM	CO+ PPM	NO++ PPM	WEIGHTED BHP	BSHC G/HP HR	BSCO+ G/HP HR	BSNO2++ G/HP HR	HUM. GR/LB
1	780	615	83	0.00	R	R	R	118.9
2	780	589	82	.07	72.16	108.63	24.95	118.9
3	680	585	166	.87	4.95	8.49	3.95	118.9
4	620	470	343	1.73	2.27	3.43	4.11	118.9
5	600	345	588	2.60	1.46	1.67	4.67	118.9
6	600	388	848	3.47	1.09	1.41	5.04	111.1
7	800	615	68	0.00	R	R	R	111.1
8	440	462	536	3.75	1.02	2.14	4.07	111.1
9	560	681	379	2.80	1.74	4.21	3.84	111.1
10	800	1244	201	1.88	3.67	11.38	3.02	111.1
11	2680	2193	74	.94	24.70	40.26	2.25	111.1
12	5680	2450	53	.10	491.06	422.04	14.93	110.0
13	780	483	66	0.00	R	R	R	110.0

CYCLE COMPOSITE
BSHC = 6.282 GRAM/BHP HR
BSCO+ = 8.757 GRAM/BHP HR
BSNO2++= 4.336 GRAM/BHP HR
BSHC + BSNO2++= 10.618 GRAM/BHP HR
BSFC = .633 LB/BHP HR

+ CONVERTED TO WET BASIS
++ CONVERTED TO WET BASIS,
CORRECTED TO 75 GRAINS OF WATER PER LB. OF DRY AIR
AND CORRECTED TO 85 DEG. F INLET TEMP. PER
FEDERAL REGISTER PARA. 85.974-18

TABLE E-3
13-MODE FEDERAL DIESEL EMISSION CYCLE

JOHN DEERE 4239T ENGINE WITH TURBOCHARGER 25 JULY 1978
PROJECT 11-5214-001 13-MODE BASELINE TEST

MODE	ENGINE SPEED RPM	TORQUE LB-FT	POWER BHP	FUEL FLOW LB/MIN	AIR FLOW LB/MIN	EXHAUST FLOW LB/MIN	FUEL AIR RATIO
1	821	0.0	0.0	.03	2.62	2.65	.013
2	1700	3.9	1.3	.08	6.08	6.16	.013
3	1700	52.0	16.8	.15	6.38	6.53	.024
4	1700	104.1	33.7	.25	6.77	7.02	.036
5	1700	156.2	50.6	.34	7.24	7.58	.047
6	1700	207.5	67.1	.56	10.77	11.33	.052
7	832	0.0	0.0	.04	2.68	2.72	.015
8	2500	186.4	88.8	.65	13.48	14.13	.048
9	2500	139.8	66.6	.50	12.55	13.05	.040
10	2500	93.2	44.4	.39	11.38	11.77	.034
11	2500	47.3	22.5	.27	10.27	10.54	.027
12	2500	3.9	1.9	.17	9.34	9.51	.018
13	822	0.0	0.0	.03	2.74	2.77	.012

MODE	HC PPM	CO+ PPM	NO++ PPM	WEIGHTED BHP	BSHC G/HP HR	BSCO+ G/HP HR	BSNO2++ G/HP HR	HUM. GR/L
1	520	477	103	0.00	R	R	R	109.7
2	520	440	80	.10	33.17	55.96	16.72	109.7
3	440	358	275	1.35	2.25	3.66	4.61	109.7
4	360	198	618	2.70	.99	1.09	5.56	109.7
5	360	166	1202	4.05	.71	.65	7.78	119.7
6	200	272	2020	5.37	.45	1.21	14.72	119.7
7	580	504	105	0.00	R	R	R	119.7
8	160	274	1635	7.10	.34	1.15	11.24	119.7
9	100	141	985	5.32	.26	.73	8.34	119.7
10	520	182	479	3.55	1.82	1.27	5.49	119.7
11	320	278	263	1.80	1.98	3.43	5.32	119.7
12	440	529	91	.15	29.44	70.56	19.98	119.7
13	560	450	105	0.00	R	R	R	119.7

CYCLE COMPOSITE
BSHC = 1.161 GRAM/BHP HR
BSCO+ = 1.995 GRAM/BHP HR
BSNO2++= 9.281 GRAM/BHP HR
BSHC + BSNO2++= 10.442 GRAM/BHP HR
BSFC = .526 LB/BHP HR

\+ CONVERTED TO WET BASIS
++ CONVERTED TO WET BASIS,
CORRECTED TO 75 GRAINS OF WATER PER LB. OF DRY AIR
AND CORRECTED TO 85 DEG. F INLET TEMP. PER
FEDERAL REGISTER PARA. 85.974-18

TABLE E-4

13-MODE FEDERAL DIESEL EMISSION CYCLE

JOHN DEERE 4239T ENGINE WITH AERODYNE TURBO   JAN 19, 1979
PROJECT 11-5214-001   NOZZLE POSITION - ZERO DEGREES

MODE	ENGINE SPEED RPM	TORQUE LB-FT	POWER BHP	FUEL FLOW LB/MIN	AIR FLOW LB/MIN	EXHAUST FLOW LB/MIN	FUEL AIR RATIO
1	858	0.0	0.0	.03	2.77	2.80	.012
2	1700	3.9	1.3	.09	5.31	5.40	.016
3	1700	52.1	16.9	.16	5.57	5.73	.028
4	1700	104.4	33.8	.24	5.97	6.21	.041
5	1700	156.5	50.7	.34	6.55	6.89	.052
6	1700	208.0	67.3	.44	7.23	7.67	.061
7	854	0.0	0.0	.04	2.61	2.65	.015
8	2500	186.8	88.9	.63	11.59	12.22	.054
9	2500	140.1	66.7	.50	10.55	11.05	.047
10	2500	93.4	44.4	.38	9.46	9.84	.040
11	2500	47.4	22.6	.26	8.21	8.47	.032
12	2500	3.9	1.9	.19	7.62	7.81	.025
13	839	0.0	0.0	.03	2.61	2.64	.013

MODE	HC PPM	CO+ PPM	NO++ PPM	BSHC G/HP HR	BSCO+ G/HP HR	BSNO2++ G/HP HR	HUM GR/LB
1	648	502	71	R	R	R	95.9
2	752	716	94	42.01	79.71	17.36	95.9
3	544	727	322	2.44	6.49	4.72	95.9
4	424	206	666	1.03	1.00	5.29	95.9
5	440	167	1347	.79	.60	7.92	95.9
6	260	416	1515	.39	1.25	7.46	95.9
7	576	443	156	R	R	R	95.9
8	178	370	1580	.32	1.34	9.38	95.9
9	134	157	1021	.29	.69	7.31	95.9
10	316	206	511	.92	1.20	4.89	95.9
11	440	428	289	2.18	4.23	4.70	95.9
12	2272	1838	45	124.90	201.40	8.17	95.9
13	736	445	59	R	R	R	95.9

CYCLE COMPOSITE   BSHC = 1.590 GRAM/BHP HR
BSCO+ = 2.874 GRAM/BHP HR
BSNO2++= 7.283 GRAM/BHP HR
BSHC + BSNO2++= 8.873 GRAM/BHP HR
BSFC = .503 LB/BHP HR

\+ CONVERTED TO WET BASIS
++ CONVERTED TO WET BASIS
CORRECTED TO 75 GRAINS OF WATER PER LB OF DRY AIR
AND CORRECTED TO 85 DEG F INLET TEMP PER
FEDERAL REGISTER PARA 85.974-18

TABLE E-5

13-MODE FEDERAL DIESEL EMISSION CYCLE

JOHN DEERE 4239T ENGINE WITH AERODYNE TURBO JAN 18, 1979
PROJECT 11-5214--001 NOZZLE POSITION - +10 DEGREES

MODE	ENGINE SPEED RPM	TORQUE LB-FT	POWER BHP	FUEL FLOW LB/MIN	AIR FLOW LB/MIN	EXHAUST FLOW LB/MIN	FUEL AIR RATIO
1	863	0.0	0.0	.03	2.91	2.94	.011
2	1700	3.9	1.3	.09	5.99	6.08	.014
3	1700	52.1	16.9	.16	6.41	6.57	.024
4	1700	104.4	33.8	.24	7.16	7.40	.034
5	1700	156.5	50.7	.33	8.15	8.48	.041
6	1700	208.0	67.3	.43	9.17	9.60	.046
7	860	0.0	0.0	.04	3.07	3.11	.012
8	2500	196.8	88.9	.63	14.21	14.84	.044
9	2500	140.1	66.7	.50	13.45	13.95	.037
10	2500	93.4	44.4	.39	12.40	12.79	.032
11	2500	47.4	22.6	.28	10.86	11.14	.026
12	2500	3.9	1.9	.17	9.27	9.44	.019
13	845	0.0	0.0	.03	2.69	2.72	.012

MODE	HC PPM	CO+ PPM	NO++ PPM	BSHC G/HP HR	BSCO+ G/HP HR	BSNO2++ G/HP HR	HUM GR/LB
1	540	417	72	R	R	R	94.8
2	528	400	108	33.21	50.18	22.34	94.8
3	432	301	313	2.22	3.08	5.28	94.8
4	340	163	655	.98	.94	6.20	94.8
5	292	116	1184	.65	.51	8.57	94.8
6	170	115	1648	.32	.43	10.15	94.8
7	540	416	96	R	R	R	94.8
8	118	193	1929	.26	.85	13.91	94.8
9	80	106	1127	.22	.58	10.19	94.8
10	190	150	525	.72	1.14	6.54	94.8
11	312	239	269	2.03	3.11	5.74	94.8
12	496	469	104	32.97	62.20	22.68	94.8
13	536	416	94	R	R	R	94.8

CYCLE COMPOSITE BSHC = 1.007 GRAM/BHP HR
BSCO+ = 1.611 GRAM/BHP HR
BSNO2++= 9.767 GRAM/BHP HR
BSHC + BSNO2++= 10.773 GRAM/BHP HR
BSFC = .502 LB/BHP HR

\+ CONVERTED TO WET BASIS
++ CONVERTED TO WET BASIS
CORRECTED TO 75 GRAINS OF WATER PER LB OF DRY AIR
AND CORRECTED TO 85 DEG F INLET TEMP PER
FEDERAL REGISTER PARA 85.974-18

TABLE E-6

RESULTS OF SMOKE TESTS WITH STANDARD (AIRESEARCH) AND AERODYNE TURBOCHARGERS AT VARIOUS LOADS AND SPEEDS

John Deere 4239T Engine

Speed, RPM	1000			1500			2000			2500		
	Std.	Aerodyne		Std.	Aerodyne		Std.	Aerodyne		Std.	Aerodyne	
	---	0°	+10°	---	0°	+10°	---	0°	+10°	---	0°	+10°
Load Lb.		174	178		187	192		181	173		156	147
Smoke, %		36.0	25.0		7.0	2.5		3.0	.5		2.5	1.5
A/F		11.78	13.29		15.32	17.85		16.89	21.19		17.19	21.73
Load Lb.	124	124	124	106	106	106	140	140	140	148	148	148
Smoke, %	6.0	9.0	6.5	1.5	2.0	1.0	1.5	2.0	.5	1.5	2.5	1.5
A/F	17.49	16.7	18.01	22.49	21.25	24.40	20.43	18.78	23.45	20.7	17.82	21.41
Load Lb	93	93	93	79.5	71.5	79.5	105	105	105	111	111	111
Smoke, %	2.5	4.0	2.5	1.5	2.0	1.0	1.0	1.5	.5	1.5	1.5	.5
A/F	23.06	21.77	23.04	27.75	26.98	28.13	20.34	21.85	27.02	23.65	20.42	25.45
Load Lb	62	62	62	53	53	53	70.5	70.5	70.5	73.5	73.5	73.5
Smoke,%	1.0	2.5	.5	1.0	1.5	.5	1.0	1.5	.5	1.5	1.5	.5
A/F	32.19	30.25	29.68	33.75	31.22	34.77	29.67	26.89	32.48	27.87	24.70	30.82

APPENDIX F

MATH MODEL DEVELOPMENT

TABLE F-1 - Mathematical Model Predictions
Cat 3208 DI NA

13-MODE FEDERAL DIESEL EMISSION CYCLE

MODE NO	SPEED RPM	TORQUE LB-FT	POWER BHP	FUEL LB/MIN	AIR LB/MIN	F/A RATIO	P4 PSIA	EXE B/LB
1	645	3.5	.4	.032	7.70	.004	21.4	18.8
2	1680	10.5	3.4	.149	19.64	.008	24.7	31.3
3	1680	119.0	38.1	.294	19.43	.015	32.1	62.0
4	1650	238.1	76.2	.467	19.15	.024	41.0	103.5
5	1680	357.1	114.2	.657	18.83	.035	51.1	154.9
6	1680	476.2	152.3	.871	18.47	.047	63.0	219.1
7	645	8.8	1.1	.034	7.70	.004	21.7	20.0
8	2800	367.6	196.0	1.289	30.49	.042	58.3	193.1
9	2800	273.1	145.6	1.010	30.98	.033	48.9	143.4
10	2800	183.8	98.0	.772	31.37	.025	41.2	104.6
11	2800	91.0	48.5	.543	31.72	.017	33.9	70.5
12	2800	7.0	3.7	.348	32.02	.011	27.9	44.2
13	650	3.5	.4	.032	7.76	.004	21.4	18.9

STOP

ENGINE MAKE AND MODEL	:CAT 3208 DI NA
NUMBER OF CYLINDERS	:8
BORE DIAMETER (IN)	:4.5
LENGTH OF STROKE	:5.
COMPRESSION RATIO	:16.5
FUEL HEATING VALUE (BTU/LB)	:18680.
SPECIFIC GRAVITY OF FUEL	:.849
STOCHIOMETRIC F/A RATIO	:.0666
COOLANT TEMPERATURE (R)	:650.
INTAKE VALVE DIAMETER (IN)	:2.09
INTAKE VALVE LIFT (IN)	:.52
ATMOSPHERIC PRESSURE (PSIA)	:14.7
ATMOSPHERIC TEMPERATURE (R)	:540.

TABLE F-2 - Mathemathical Model Predictions
Hino Model EH700E

13-MODE FEDERAL DIESEL EMISSION CYCLE

MODE NO	SPEED RPM	TORQUE LB-FT	POWER BHP	FUEL LB/MIN	AIR LB/MIN	F/A RATIO	P4 PSI	EXE B/LB
1	541	0.0	0.0	.014	4.00	.003	20.7	16.5
2	2000	7.0	2.7	.112	14.40	.008	24.8	31.8
3	2000	77.0	29.3	.223	14.23	.016	32.2	62.9
4	2000	150.6	57.3	.348	14.02	.025	40.7	103.0
5	2000	225.8	86.0	.489	13.79	.035	50.7	153.6
6	2000	297.6	113.3	.640	13.54	.047	61.7	213.4
7	540	0.0	0.0	.014	4.00	.003	20.7	16.5
8	3000	250.3	143.0	.896	20.10	.045	59.2	199.4
9	3000	187.3	107.0	.699	20.42	.034	49.5	147.5
10	3000	127.8	73.0	.530	20.70	.026	41.5	106.6
11	3000	61.3	35.0	.357	20.98	.017	33.4	68.5
12	3000	7.0	4.0	.224	21.20	.011	27.4	42.3
13	534	0.0	0.0	.013	3.95	.003	20.7	16.4

STOP

ENGINE MAKE AND MODEL	:HINO MODEL EH700E
NUMBER OF CYLINDERS	:6
BORE DIAMETER (IN)	:4.33
LENGTH OF STROKE	:4.45
COMPRESSION RATIO	:18.2
FUEL HEATING VALUE (BTU/LB)	:18660.
SPECIFIC GRAVITY OF FUEL	:.849
STOCHIOMETRIC F/A RATIO	:.0666
COOLANT TEMPERATURE (R)	:650.
INTAKE VALVE DIAMETER (IN)	:1.89
INTAKE VALVE LIFT (IN)	:.49
ATMOSPHERIC PRESSURE (PSIA)	:14.7
ATMOSPHERIC TEMPERATURE (R)	:540.

F.4

TABLE F-3 - Experimentally Determined Fuel and Air Flow Rates

13-MODE FEDERAL DIESEL EMISSION CYCLE

PROJECT: 11-4909-002 TEST DATE 4-7-78 TEST NO.3 994 HRS
ENGINE: CATERPILLAR 3208 DI NA SERIAL NO.1A6385

MODE	ENGINE SPEED RPM	TORQUE LB-FT	POWER BHP	FUEL FLOW LB/MIN	AIR FLOW LB/MIN	EXHAUST FLOW LB/MIN	FUEL AIR RATIO
1	645	3.5	.4	.04	7.08	7.12	.006
2	1680	10.5	3.4	.14	19.44	19.58	.007
3	1680	119.0	38.1	.27	19.42	19.69	.014
4	1680	238.1	76.2	.44	19.30	19.74	.023
5	1680	357.1	114.2	.63	18.88	19.51	.033
6	1680	476.2	152.3	.91	18.39	19.29	.049
7	645	8.8	1.1	.04	7.04	7.08	.006
8	2800	367.6	196.0	1.34	28.19	29.53	.047
9	2800	273.1	145.6	.97	28.49	29.46	.034
10	2800	183.8	98.0	.75	28.51	29.26	.026
11	2800	91.0	48.5	.50	28.22	28.72	.018
12	2800	7.0	3.7	.31	28.10	28.42	.011
13	650	3.5	.4	.04	7.12	7.17	.006

TABLE F-4 - Experimentally Determined Fuel and Air Flow Rates

13-MODE FEDERAL DIESEL EMISSION CYCLE

PROJECT: 11-4978-002 TEST DATE 4-6-78 TEST NO.2 130 HRS
ENGINE: HINO MODEL EH700E SERIAL NO.32397

MODE	ENGINE SPEED RPM	TORQUE LB-FT	POWER BHP	FUEL FLOW LB/MIN	AIR FLOW LB/MIN	EXHAUST FLOW LB/MIN	FUEL AIR RATIO
1	541	0.0	0.0	.03	3.97	4.00	.007
2	2000	7.0	2.7	.10	15.61	15.71	.006
3	2000	77.0	29.3	.23	15.41	15.64	.015
4	2000	150.6	57.3	.37	15.24	15.61	.024
5	2000	225.8	86.0	.52	15.04	15.55	.034
6	2000	297.6	113.3	.71	14.81	15.52	.048
7	540	0.0	0.0	.02	3.98	4.00	.006
8	3000	250.3	143.0	1.00	21.21	22.22	.047
9	3000	187.3	107.0	.76	21.61	22.37	.035
10	3000	127.8	73.0	.56	21.64	22.21	.026
11	3000	61.3	35.0	.38	21.64	22.02	.018
12	3000	7.0	4.0	.24	21.59	21.83	.011
13	534	0.0	0.0	.03	3.92	3.94	.007

TABLE F-5

MODEL ESTIMATION OF HIGHWAY FUEL ECONOMY

```
ENGINE MAKE AND MODEL                   :VW DIESEL
NUMBER OF CYLINDERS                     :4
BORE DIAMETER (IN)                      :3.012
LENGTH OF STROKE                        :3.15
DISPLACEMENT (CU IN)                    :89.77829990976
COMPRESSION RATIO                       :23.
INTAKE VALVE DIAMETER (IN)              :1.338
INTAKE VALVE LIFT (IN)                  :.3
ATMOSPHERIC PRESSURE (PSIA)             :14.6
ATMOSPHERIC TEMPERATURE (R)             :540.
COOLANT TEMPERATURE (R)                 :660.
FUEL HEATING VALUE (BTU/LB)             :18680.
SPECIFIC GRAVITY OF FUEL                :.849
STOCHIOMETRIC F/A RATIO                 :.0666
FRONTAL AREA (SQ FT)                    :20.
WEIGHT OF THE VEHICLE (LB)              :2250.
          FUEL CONSUMED (GMS)               =616.9996496545
          FUEL ECONOMY (MPG)                =53
          IDLE PERIOD (SEC)                 =3
          BRAKING PERIOD (SEC)              =0
          0-4.74 MPH PERIOD (SEC)           =9
          4.74-15.0 MPH PERIOD (SEC)        =7
          15-25 MPH PERIOD (SEC)            =9
          25-40 MPH PERIOD (SEC)            =72
          ABOVE 40 MPH PERIOD (SEC)         =667
  STOP
```

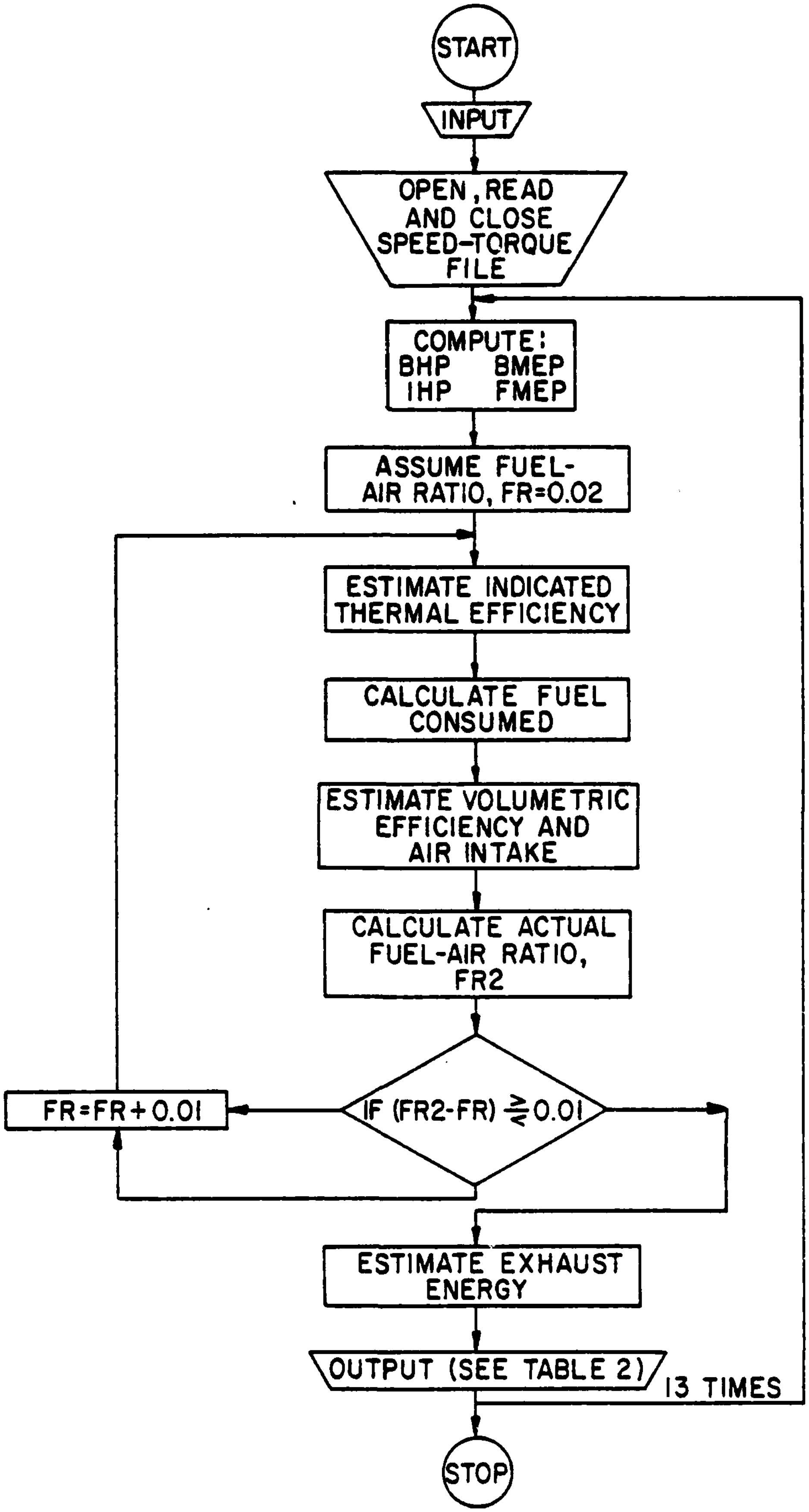


FIGURE F-1 - MATH MODEL FLOW CHART FOR A NATURALLY ASPIRATED DIESEL ENGINE OVER THE 13 MODE FEDERAL DIESEL EMISSION CYCLE

AD-A081 442 AERODYNE DALLAS TX F/G 21/4
TURBOCHARGING OF SMALL INTERNAL COMBUSTION ENGINES AS A MEANS O--ETC(U)
1979 DAAK70-78-C-0031

UNCLASSIFIED NL

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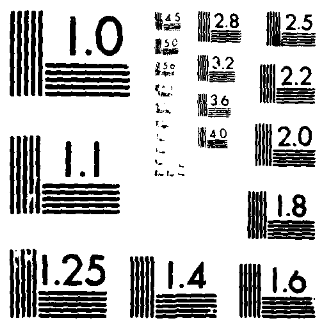

MICROCOPY RESOLUTION TEST CHART
NATIONAL BUREAU OF STANDARDS-1963-A

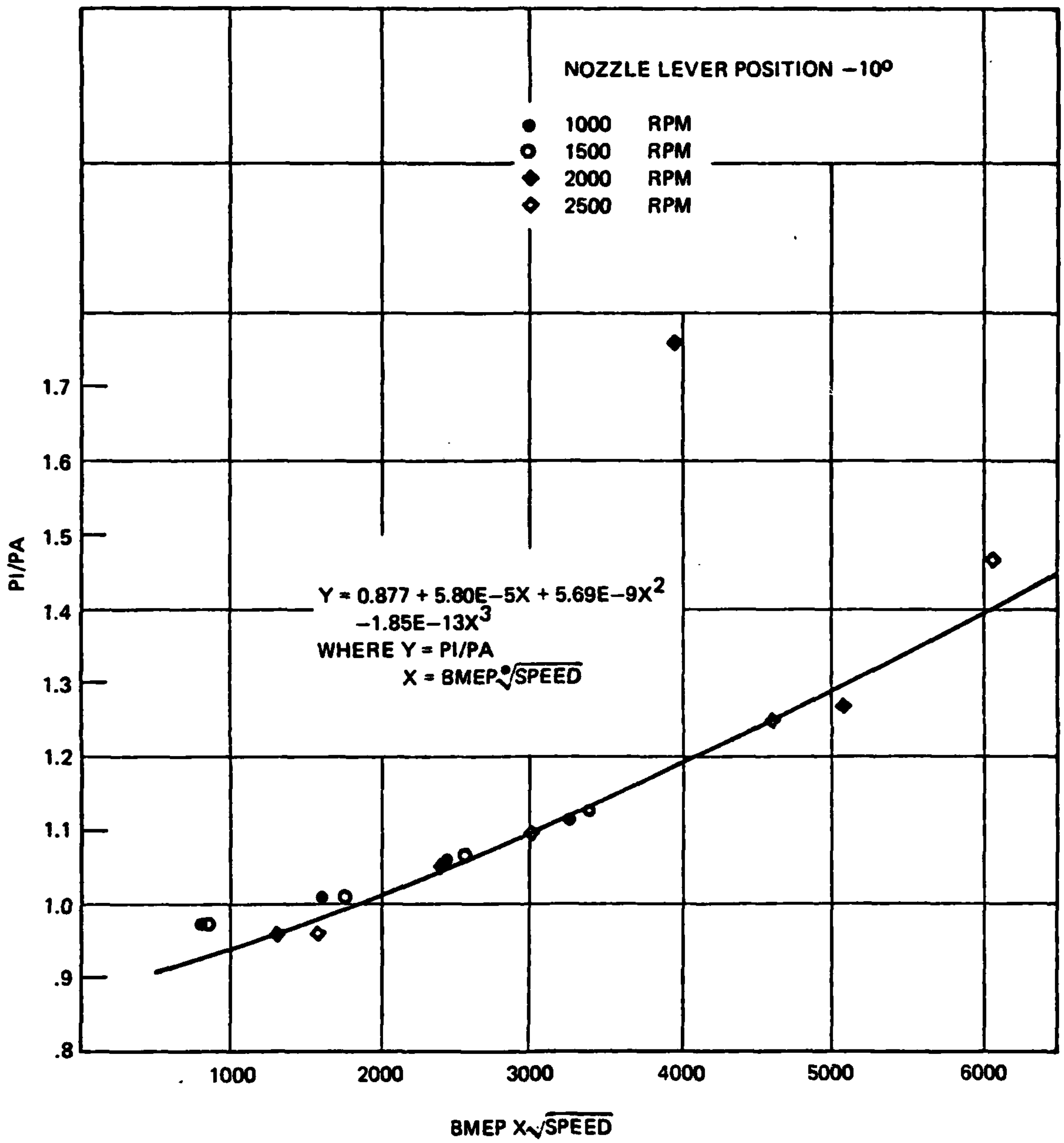

FIGURE F-2 - PRESSURE BOOST AS A FUNCTION OF BMEP AND ENGINE SPEED - 10° NOZZLE POSITION

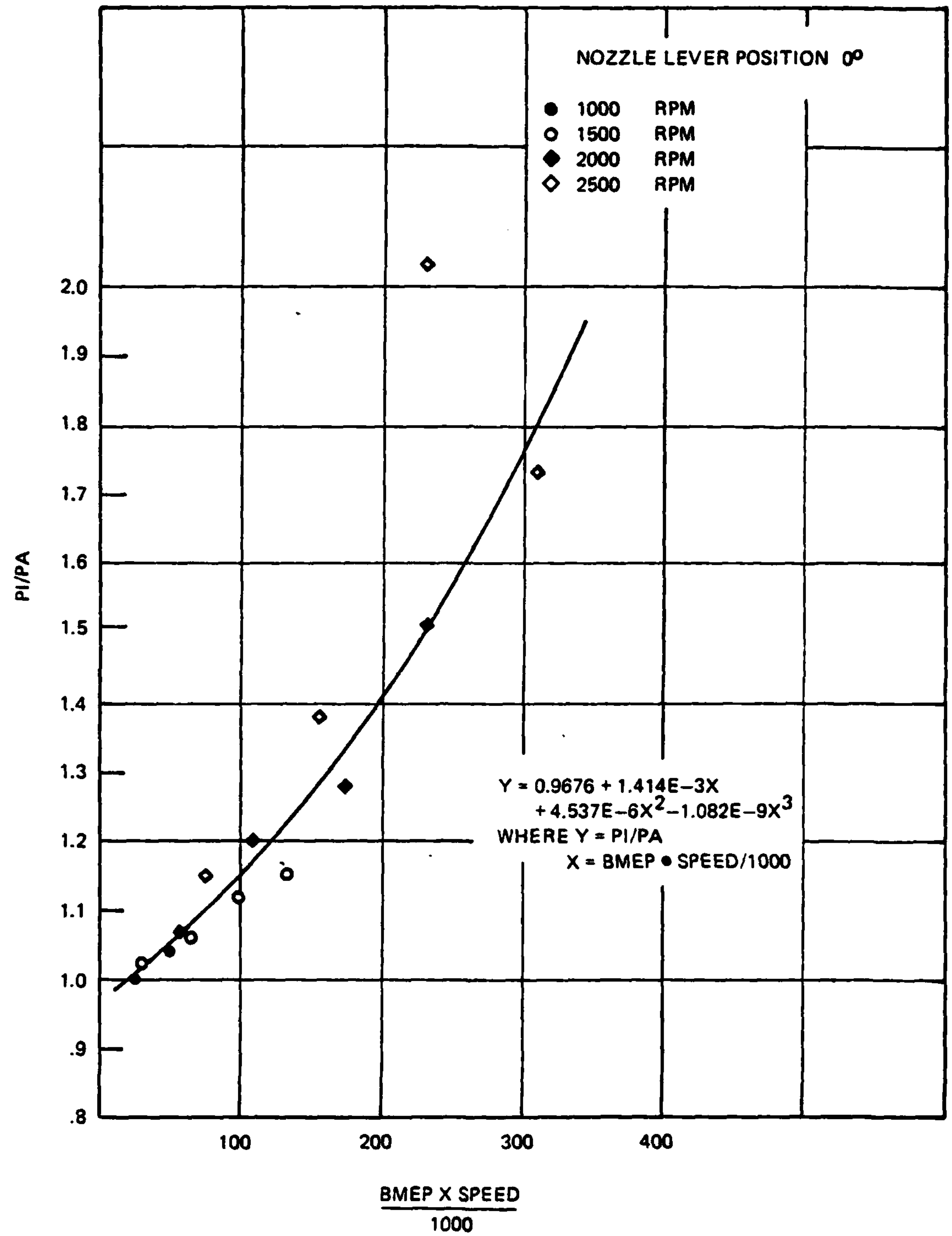

FIGURE F-3 - PRESSURE BOOST AS A FUNCTION OF BMEP AND ENGINE SPEED - 0° NOZZLE POSITION

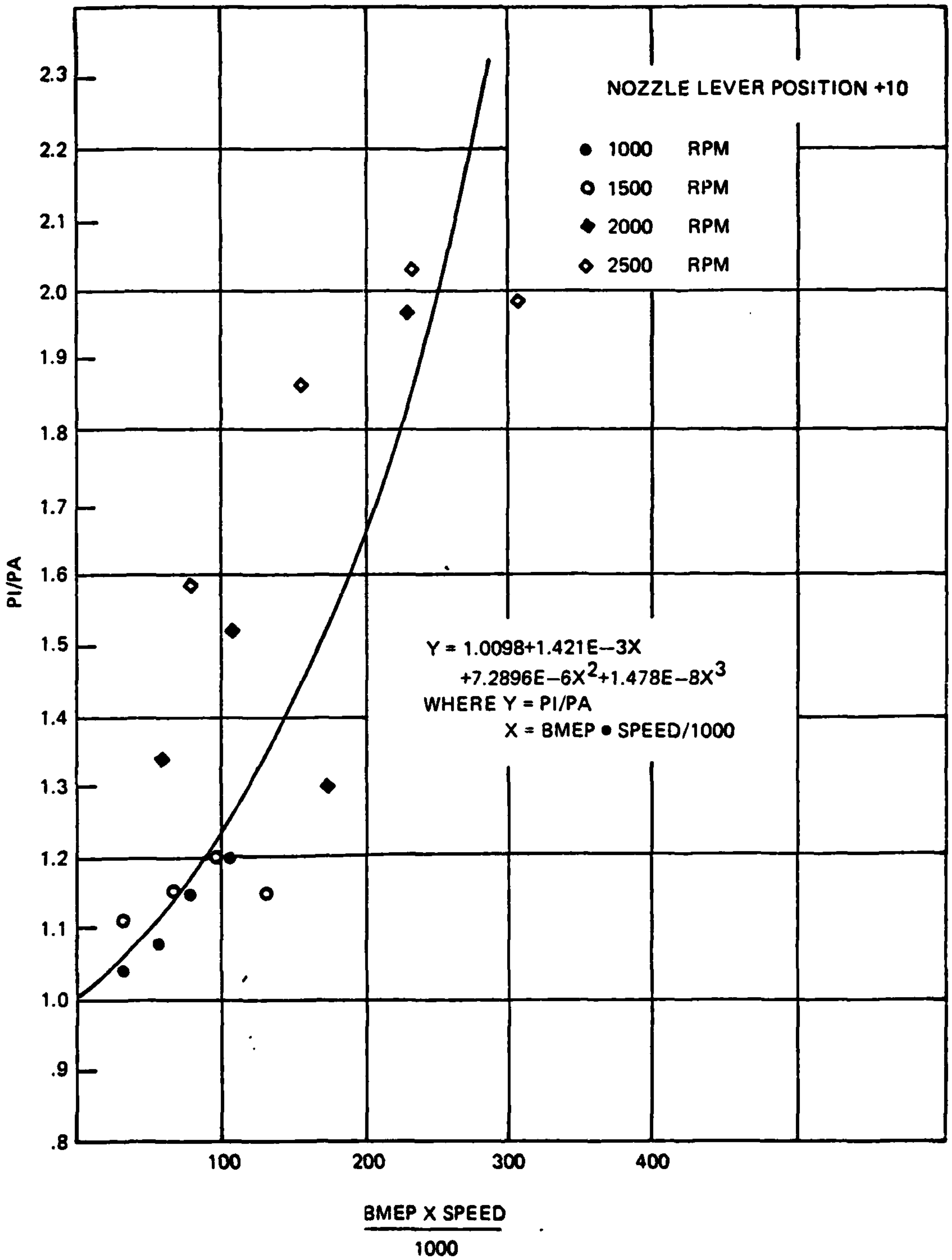

FIGURE F-4 - PRESSURE BOOST AS A FUNCTION OF BMEP AND ENGINE SPEED - + 10° NOZZLE POSITION

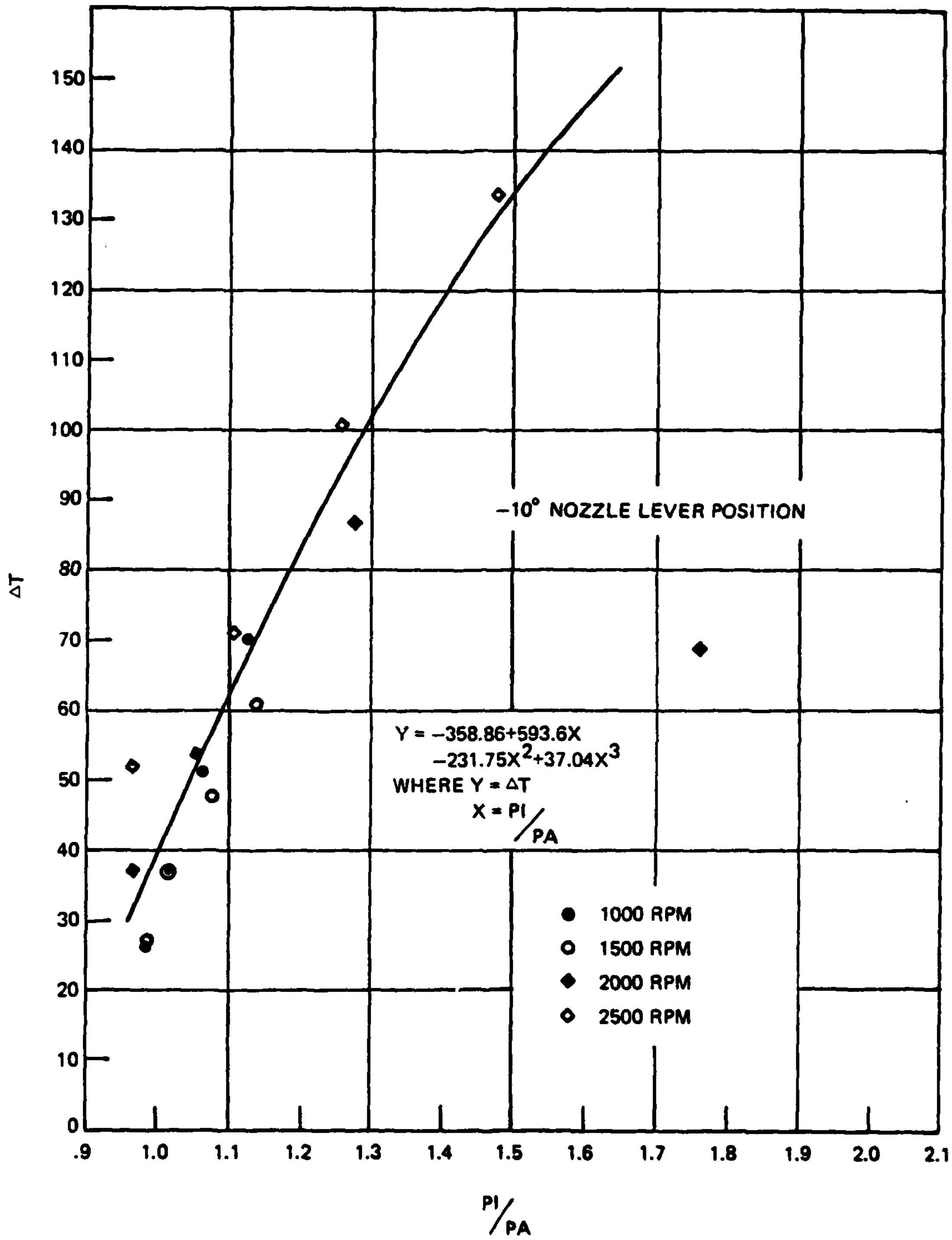

FIGURE F-5 - TEMPERATURE RISE ACROSS THE COMPRESSOR AS A FUNCTION OF BOOST PRESSURE - -10° NOZZLE POSITION

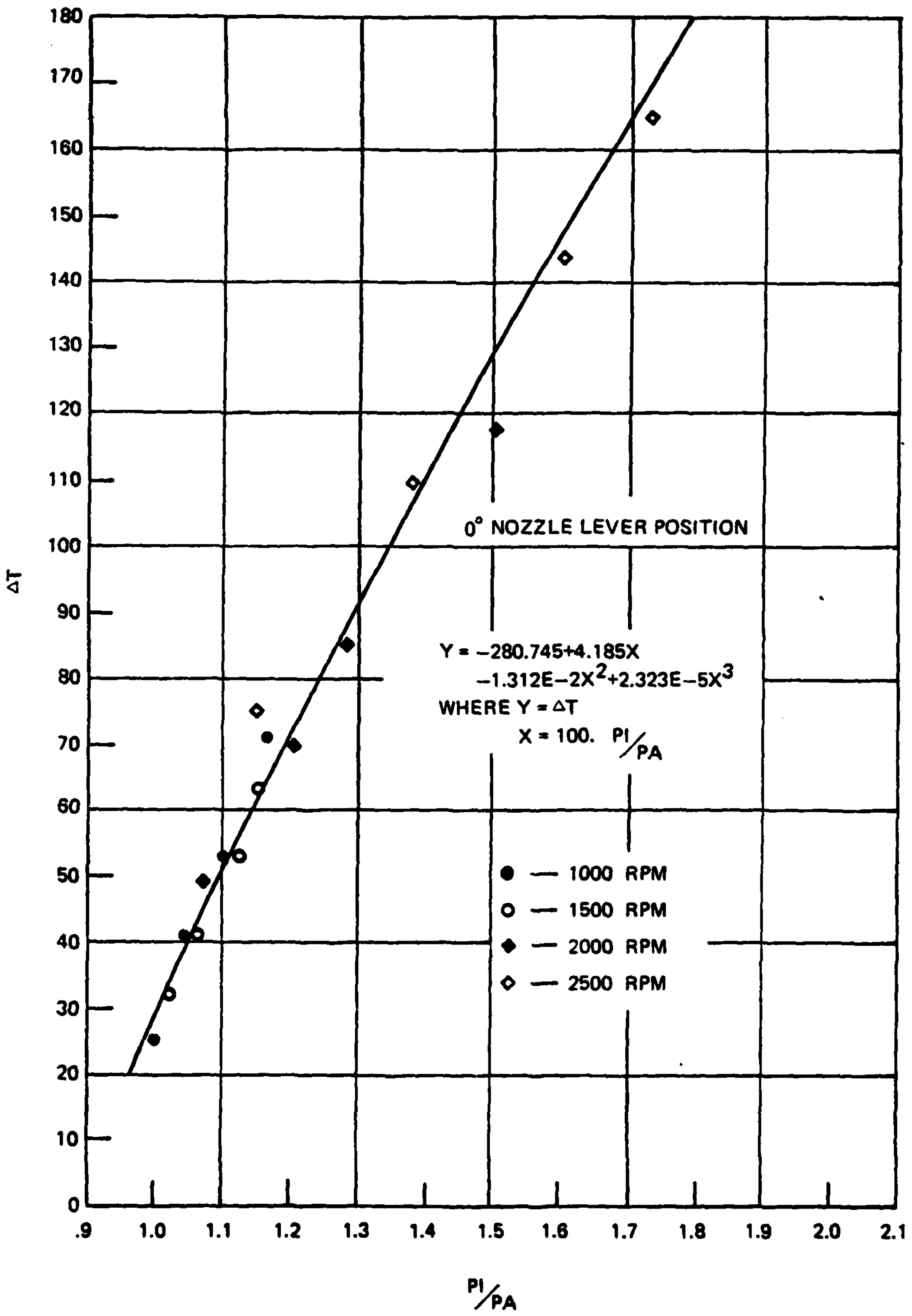

FIGURE F-6 - TEMPERATURE RISE ACROSS THE COMPRESSOR AS A FUNCTION OF BOOST PRESSURE - 0° NOZZLE POSITION

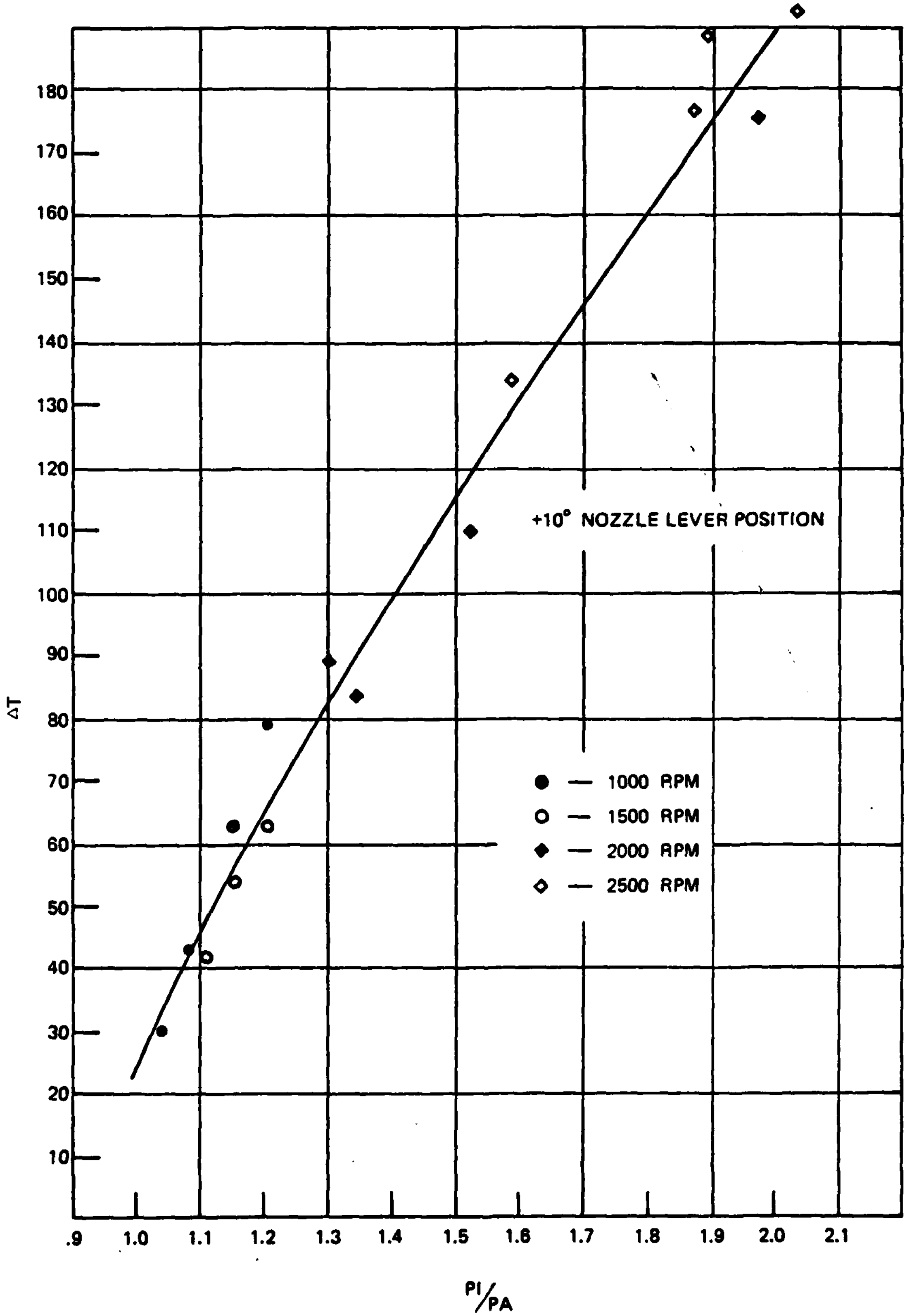

FIGURE F-7 - TEMPERATURE RISE ACROSS THE COMPRESSOR AS A FUNCTION OF BOOST PRESSURE - +10° NOZZLE POSITION

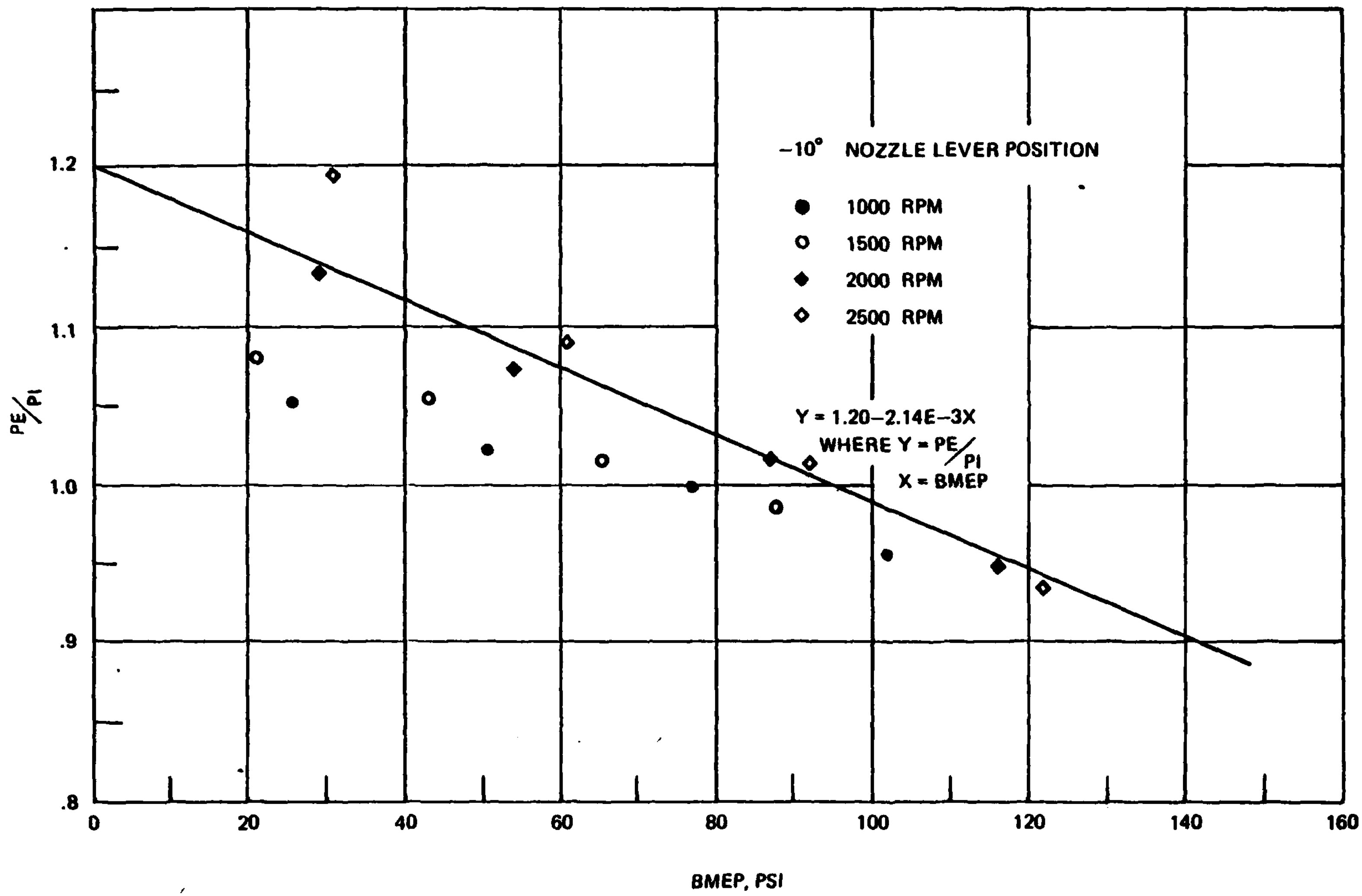

FIGURE F-8 - EXHAUST TO INTAKE MANIFOLD PRESSURE RATIO AS A FUNCTION OF BMEP - -10° NOZZLE POSITION

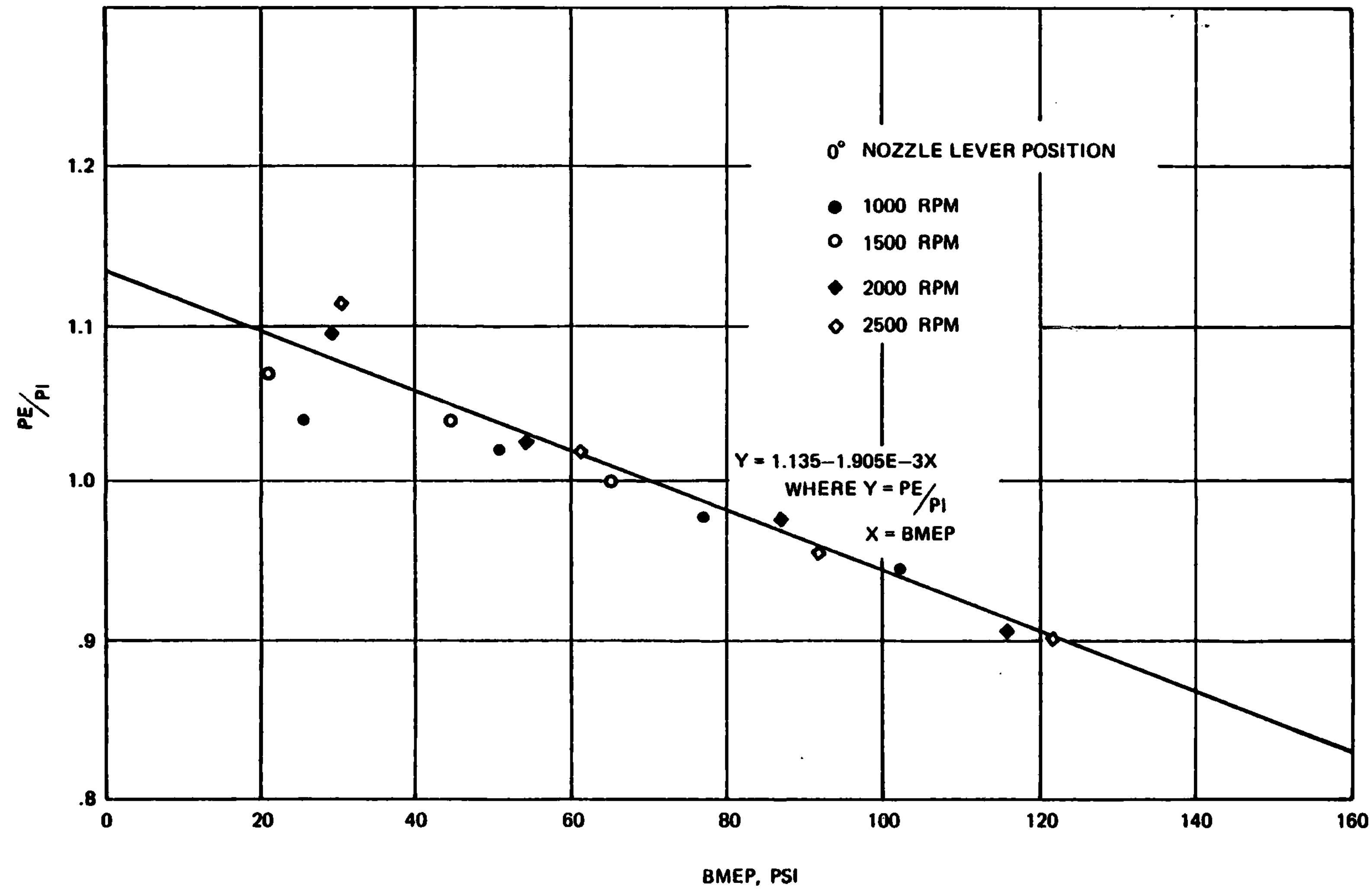

FIGURE F-9 - EXHAUST TO INTAKE MANIFOLD PRESSURE RATIO AS A FUNCTION OF BMEP - 0° NOZZLE POSITION

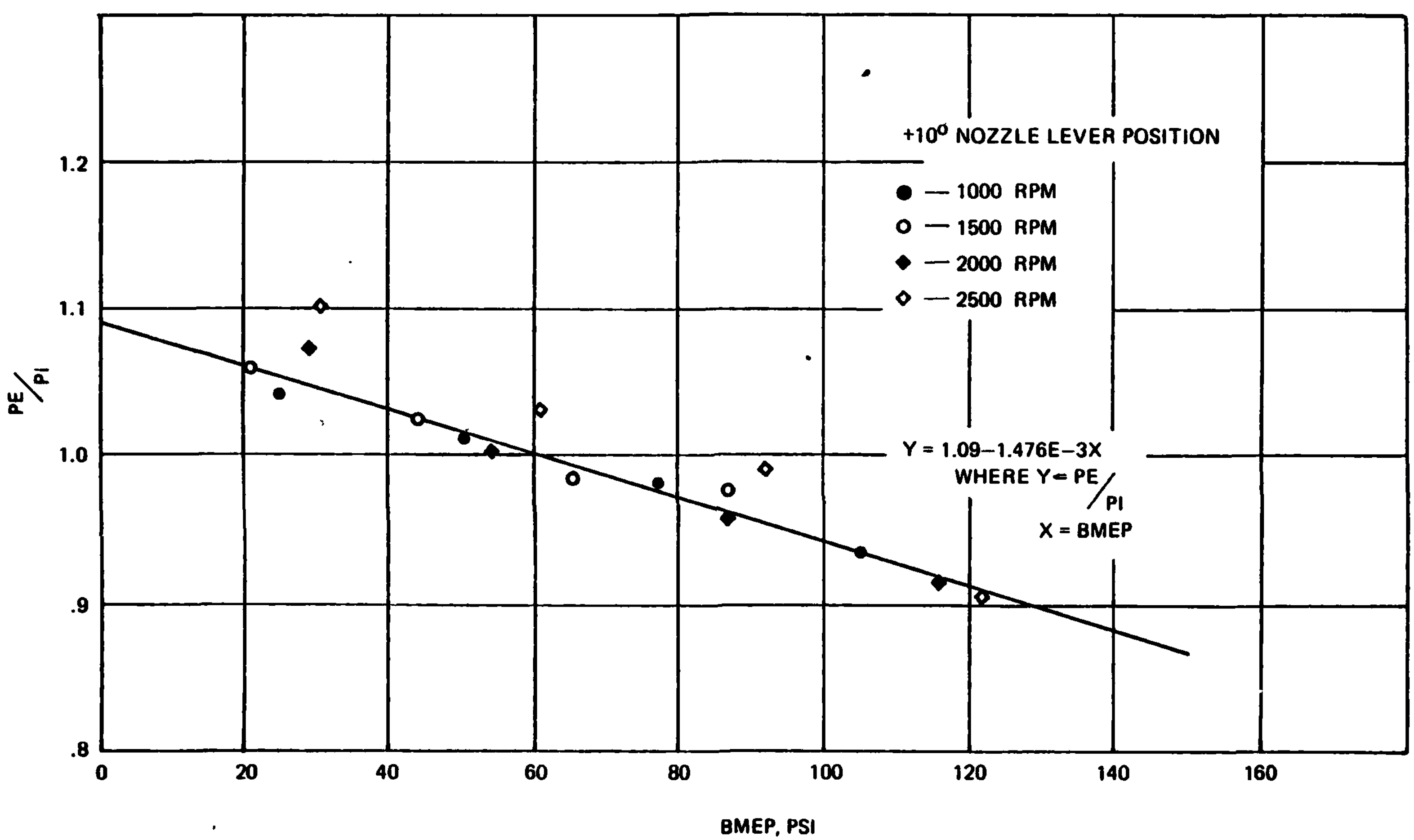

FIGURE F-10 - EXHAUST TO INTAKE MANIFOLD PRESSURE RATIO AS A FUNCTION OF BMEP - +10° NOZZLE POSITION